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WATER-TUBE *VS.* CYLINDRICAL BOILERS FOR NAVAL VESSELS.

BY W. LEDYARD CATHCART, MEMBER.

The water-tube boiler has a long history, whose later years have been eventful. The honor as to the first recorded invention of this type is claimed for John Blakey, whose boiler is said to have been patented in 1774. Perhaps the most noteworthy among early installations in steam vessels was that of Colonel John Stevens, who, in 1804, fitted a water-tube boiler in a small steamboat on the Hudson River. From the beginning of the nineteenth century onward, the advantages of the type, when fitly designed for the conditions to be met, have stirred the interest of inventors, and their work has been unceasing.

With regard to marine service, however, it is scarcely fifteen years since boilers of this type began to win their way into extensive use. The forerunner of modern naval installations appeared in 1856, in the fitting of the Belleville boiler of that day in the French war vessel *Biche*. Three years later there were conducted at the Navy Yard, New York, comparative tests of the horizontal fire-tube boiler and of one with vertical water tubes designed by Engineer-in-Chief D. B. Martin, U. S. Navy. It is

interesting to note that the much higher evaporative efficiency of the Martin boiler and the ease with which it could be scaled led to its almost universal use in our naval vessels. The tubes were, however, surrounded by a shell of rectangular section, and the boiler was therefore unsuitable for high pressures.

With the advent of the compound engine and the growth of steam pressures, the marine cylindrical boiler, in the absence of any formidable competitor of water-tube type, began a long career of usefulness which met no serious check until about fifteen years ago, and which now, in naval vessels at least, seems about to close. It is true, that during the period of its supremacy, the progress of invention with the water-tube type continued with increasing installations in launches, yachts and occasional merchant steamers. It was not until 1888, however, that the French naval authorities took radical action in deciding to abandon the cylindrical type, to extend the use of the Belleville boiler, then fitted in the despatch vessel *Le Voltigeur* and a cruiser, and to begin a number of installations of the d'Allest type. In 1885 Great Britain made its first naval test of the water-tube type in the Thornycroft boiler as applied in a torpedo boat, which boiler was also fitted in 1887 in H. M. S. *Speedy*, of 4,500 I.H.P. In 1893 the decision to equip the cruisers *Powerful* and *Terrible*, each of 25,000 I.H.P., with the Belleville boiler was the first important step toward the adoption in 1894 of the Belleville type for all new vessels of large displacement. The water-tube boiler re-appeared in the U. S. Navy in those of Thornycroft design installed in the torpedo boat *Cushing*, launched in 1890, and in the Ward boilers fitted, in conjunction with the cylindrical type, in the monitor *Monterey* in 1892. The Engineer-in-Chief, in his report for the year 1897, announced the full revival of the type in our Navy in stating that "the efficiency of the fleet will be best served by using water-tube boilers on future ships."

PRESENT STATUS OF THE WATER-TUBE BOILER.

It will be seen that, from 1888 onward, the progress of the water-tube boiler in naval service has been swift. In torpedo

craft it has long since vanquished its cylindrical competitor, and the change is complete, since the requirement in these boats of very high speed with small displacement has forced the development of the greatest possible boiler power on the least possible allowances for weight and space, while the locomotive boiler—the immediate predecessor of the water-tube in this service—was not only relatively heavy, but, under the necessary limitations as to weight and space, required an air pressure sometimes reaching 8 inches, to develop the power required. This severe forcing of a relatively rigid structure led to leakage and frequent disaster.

While the cylindrical boiler has not yet been discarded wholly in cruising vessels, its day in that field also seems to have passed. In the U. S. Navy the last battleships thus equipped were those of the *Wisconsin* class, contracted for in 1896. All subsequent installations have been of the water-tube type.

The following table gives, excluding torpedo craft, the water-tube boiler installation in U. S. naval vessels:

BATTLESHIPS AND ARMORED CRUISERS.

Niclausse design:—*Maine, Virginia, Pennsylvania, Colorado, Georgia.*

Thornycroft design:—*Missouri, Ohio.*

Babcock & Wilcox design:—*Nebraska, New Jersey, Rhode Island, Connecticut, Louisiana, Vermont, Kansas, Minnesota, West Virginia, California, Maryland, South Dakota, Tennessee, Washington.*

PROTECTED CRUISERS AND GUNBOATS.

Babcock & Wilcox design:—*Cincinnati, Raleigh, Denver, Des Moines, Chattanooga, Galveston, Tacoma, Cleveland, St. Louis, Milwaukee, Charleston.*

MONITORS AND AUXILIARY VESSELS.

Babcock & Wilcox design:—*Wyoming, Amphitrite, Marietta, Annapolis, Dubuque, Paducah.*

Hohenstein design:—*Helena, Wilmington.*

Thornycroft design:—*Arkansas.*

Niclausse design :—*Nevada.*
Mosher design :—*Florida.*

COMBINED WATER-TUBE AND SCOTCH INSTALLATIONS.

B. & W. and cylindrical design :—*Atlanta, Chicago.*
Ward and cylindrical design :—*Monterey.*
Yarrow and cylindrical design :—*Nashville.*

In the late constructions of nearly all navies similar conditions exist. Thus, the recent battleships and large cruisers, projected or building, by France, Russia, Italy, Germany, Austria, Holland, Sweden and Japan will be fitted with water-tube boilers of various types, including the Belleville, Babcock & Wilcox, Thornycroft-Schulz, Dürr, Niclausse, Normand, and Yarrow.

In the British navy, however, there has been a partial reversion to the cylindrical type, owing to the recommendations of the Boiler Committee appointed by the Admiralty. In its report of May 28, 1902, this committee stated:

"Experience thus obtained has confirmed the Committee in the opinions expressed in their interim report, viz: 'that the advantages of water-tube boilers for naval purposes are so great, chiefly from a military point of view, that, providing a satisfactory type of water-tube boiler be adopted, it would be more suitable for use in H. M. Navy than the cylindrical type of boiler.' * * * Until a thoroughly satisfactory type of water-tube boiler is obtained, the Committee therefore recommend that, in large cruisers and battleships, cylindrical boilers of sufficient power to work the auxiliary machinery and to drive the ship at her customary cruising speed, should be fitted; the steam pressure should be the same for the water-tube and cylindrical boilers, and may conveniently be 210 pounds per square inch, so as to give 200 pounds at the engines."

The Committee's objection to the water-tube boiler was based on their view that it was not as economical as the cylindrical type, especially in port. It is worthy of note that this Committee was composed of an admiral, five eminent engineers from civil life, and but one naval engineer. As a consequence of this re-

port, the recent boiler installations of the British Navy comprise not only the Belleville, Babcock & Wilcox, and Niclausse types singly, but also the two latter, the Yarrow and the Dürr, each combined with cylindrical boilers having one-fifth of the total power required.

The latter policy—which has been tried and, in the latest vessels, abandoned by both the United States and Germany—will probably be but partial and temporary in Great Britain. As compared with the water-tube type, the combined installations have greater weight, less steam pressure and some difficulties in operation. These disadvantages, with the steady advance of the water-tube boiler, will settle the question. As a whole, it may be said that the cylindrical boiler is, for the time at least, practically excluded from the recent vessels of the world's navies, since even the proposed British composite installations comprise but 20 per cent. of the total power required in each combination.

As to the apparent regression of these combined installations, it may not be amiss to note the similarity in some respects between the long and bitter struggle of the water-tube boiler for full recognition in the British fleet and that of breech-loading ordnance in the same service. As far back as 1850 the Wahrendorff and Cavalli guns of this type were tested at Shoeburyness; in 1860–1, the Armstrong rifle was introduced in the Navy; in 1865 there was a reversion to muzzle-loading, and the equipment of ships with guns on the Woolwich system began. This retrograde policy reached its climax in 1880, when the wide adoption by other naval powers of breech-loading guns, which gave increased length, higher velocity and greater penetration, forced England to follow suit. Admiral O'Neil points out that our own service also was not gifted with great foresight in the early days of the breech loader, even the text book on gunnery used at the Naval Academy in 1862, stating that "In spite of the apparent success of the Armstrong and Whitworth gun, there are objections to breech-loading cannon which must be pointed out and which go to prove that the endeavor to produce breech-loading cannon is an effort to obtain uncalled for and superfluous facility in gunnery." In resisting the introduction of the water-

tube boiler in naval vessels, its opponents have been governed, unwittingly or not, by the principle that its tactical advantages gave an "uncalled for and superfluous facility" in that respect.

DIFFICULTIES MET IN THE INTRODUCTION OF THE WATER-TUBE BOILER.

The wide diversity of design in the water-tube boilers fitted in recent vessels shows that no one type yet commends itself to all, and marks one of the chief difficulties met in the introduction of the water-tube. The cylindrical boiler, although constantly improved during its long service, has been from the beginning, in broad essentials, the same in all navies. On the other hand, the water-tube boiler, reduced to its elements, is but a closed circuit whose heating surface is tubular, into which water is introduced and from which steam is withdrawn. The variations in the detailed application of this broad principle have been numberless, with consequent change in the relative merits and disadvantages of different designs. There are all kinds of water-tube boilers, ranging from those fitted with a wide variety of special appliances to correct inherent defects, to those which, in their simplicity, are but a step beyond the cylindrical boiler. Of these, none has failed purely because it was a water-tube boiler, but not a few have suffered because of defective circulation, incomplete combustion, the intricacy of automatic feeding apparatus, inacessibility for cleaning and repair, or poor workmanship. Boilers of proper design have found their adoption retarded by the sins, in these respects, of weaker brethren.

A word may be said also as to the difficulties met in the widely varying character of the evidence of the water-tube boiler's efficiency as obtained from shore tests and sea trials, mainly in war vessels. A boiler is responsible for its own efficiency only, and not for those of the fire-room force and of the engines and auxiliary machinery to which it supplies steam. The number of heat units which a pound of coal of given composition and condition will produce, when burned with full economy, is a fixed quantity of known evaporative power, and tests restricted to the boiler only, when properly conducted as to firing, the weights of water and coal, and the analysis of the latter, will give the true

efficiency of the boiler for the various rates of combustion through which the tests extended. On the other hand, in sea trials, boiler efficiency is inseparable from the additional efficiencies noted above, and the final result in "Pounds of coal per I.H.P." is often a vague and misleading measure of a boiler's value, especially if the composition of that coal and the amount of water evaporated per pound are unknown.

Intelligent firing is a factor of much importance in the aggregate efficiency of an installation including water-tube boilers. The amount of water contained is small as compared with the extent of heating surface, making steam generation a sensitive action, while the grates are wide, with two or more doors giving a sea-barbarian fine opportunities for checking combustion by a fire of hills and hollows and by the frequent inrush of cold air through doors open too often and too long. What is required is simply a fire of even and moderate depth throughout, with coal fed at regular intervals and in regular order through the various doors. Without system and a fair amount of intelligence and training in this, the efficiency of the boiler will be seriously impaired.

As to the part which the engine fills in fixing the aggregate efficiency of the installation, it is apparent that the naval engine of short stroke and high piston speed is not as economical as engines of the highest class in the merchant marine. Thus the *Kaiser Wilhelm der Grosse* at 22.79 knots develops 6.71 I.H.P. per ton of total weight of machinery, including water, while the similar figure, based on estimated weights, for the battleship *Georgia* at 19 knots is 10.98 I.H.P. per ton, *i. e.*, even excluding the weight saving from the use of water-tube boilers in the battleship, the latter, for the same weight, develops at least fifty per cent. more horsepower than is required from the liner. Again, in judging the efficiency of naval machinery at various powers, two facts should be remembered: 1st, that in U. S. naval engines the ratio of cylinder areas is not sufficient to give the best steam economy at maximum power, since the decrease in that economy would be too great at the ordinary cruising speed; 2d, that the cruising power is but one-eighth to one-tenth of

that at maximum speed. For example, the *Georgia's* maximum steam pressure is 250 pounds at the engines, but her cylinder ratio is that suitable for 160 pounds, with a corresponding speed of 15.4 knots. Steaming in squadron at 10 knots she would require about 2,400 I.H.P., or one-eighth of her maximum. Thus, at 15.4 knots her steam economy would reach its maximum, at 19 knots she would be wasting heat units in the exhaust, and at 10 knots in cylinder condensation. As to the auxiliary machinery, the noteworthy tests of Lieut. W. W. White, U. S. Navy (Vol. X, No. 1), have shown its extravagant use of steam, especially at low powers. Without experimental data, a reasonable estimate of the steam consumption for this machinery at various powers is difficult. In view of the wide variation in form, complexity, care in operation and in tests, especially in war vessels, it will be seen that, in judging the water-tube boiler as a type, the evidence, in each instance, should be analyzed with regard to the design of the boiler and the conditions of the trial.

THE CYLINDRICAL BOILER IN WAR SERVICE.

Many extravagant claims have been advanced by engineers devoted to their "old and proved friend, the Scotch boiler." In the now voluminous literature of a boiler controversy which has almost reached its majority, whose arguments, logical and the reverse, have been long since worn threadbare by age, the virtues of the cylindrical boiler in efficiency, cost of maintenance and endurance to hoary age have figured largely. These assertions have been based mainly upon experience with merchant vessels in which, steaming continuously and uniformly, the boiler passes from one port to another, is cleaned and repaired, and begins at once its return run. Under such conditions, the cylindrical boiler—barring its weight, its limitations as to pressure and forced draft, and its slowness in raising steam—has served well, although the confidence in its efficiency and economy of maintenance comes chiefly from the intimate knowledge, gained by years of use, as to its proper operation and repair.

In the irregular service of the Navy, however, it is a different story. Pressed to the utmost one day, the boiler may lie idle the

next and for months in port, or, in war, may drift, as off Santiago, for weeks under banked fires. These severe conditions make its life relatively short and increase largely the care required to maintain its efficiency. In his lectures before the Naval War College in 1892, Professor Hollis, speaking from his experience with cylindrical boilers, said: "There is no part of a cruiser so important as the boilers. The guns, torpedoes and main engines take care of themselves with a very moderate amount of attention, but the boilers are unremitting in their cries for help. When a ship is laid up, it is the boilers in nine cases out of ten. * * * In the long run she will depend for her success upon the boilers." Admiral FitzGerald, R. N., in addressing the Institution of Naval Architects in 1897, said: "My conclusion is that the Scotch and marine locomotive type of boilers have proved themselves to be unequal to the demands made upon them for the greater pressures and quantities of steam required for the working of fast-running, triple-expansion engines. It has been one long, monotonous tale of leaky tube plates and seams, bulged fire boxes and generally more or less disabled and inefficient boilers. The instrument has proved to be unsuitable to the work required of it, and it has failed." These words are none too strong in their description of British disaster in the attempt to force the cylindrical boiler to provide too large a volume of steam, even at moderate pressure, on a too limited allowance for weight and space.

No such misfortunes occurred in our Navy because our cylindrical boilers were restricted to and made sufficiently strong for the work of which they were capable. Despite this, however, the inherent defects of the system were shown conspicuously during the brief service of our naval vessels in the war with Spain. Lieutenant J. K. Robison, U. S. Navy, says (Vol. X, No. 4):

"It would appear, however, that in order to retain anything like the true efficiency of the ship, it is necessary to keep intact at all times the power of securing a full steam supply at short notice. The battle of Santiago amply demonstrated that the time allowed to get the steam to the engines is considerably less

than the time required to raise steam from a cold cylindrical boiler. In all cases where the fires were not actually lighted in the boilers they were kept in readiness for instant lighting. There may have been a few exceptions to this, which is intended to be merely a statement of a general rule. * * * Lying with all boilers in use and with the fires kept clean, so as to be ready for emergencies, with steam on the engines, required the expenditure of a large amount of coal. The number of knots made each day was small, but much coal was lost in the stacks, in radiation and through the engine drains. This 'standing by' costs heavily in coal. * * * The battleship *Massachusetts* lost her chance of sharing in the glories of July 3d because she was away 'coaling ship.' * * * The work of the men who were forced to stand steaming watches for such a long period in the intense heat of the fire rooms and engine rooms was very hard. * * * This work was well done, but the men were weakened. * * * Naturally, this decreased the power of the ships below what it should have been. * * * The use of all boilers increased the loss of feed water, because the number of leaks was increased. * * * The evaporating plants were frequently insufficient to maintain the freshness of the feed water. * * * This, of course, made necessary the use of salt feed as a 'make-up.' * * * With the boilers always in use they soon became dirty, not only on the fire side, but also on the water side of the boilers. Leaky condensers gave much trouble, and it has been seen that salt water in the boilers was a common occurrence. Now, these boilers, being always in use, were inaccessible for repairs or cleaning, and much efficiency was lost as a result of the scale that accumulated upon the heating surfaces. There are several cases recorded of the dropping of the crown sheets of furnaces from this cause. In at least one case, that of the *Indiana*, a battleship was thus disabled for a considerable time. * * * The mixed crews, with fires that soon became dirty and that never were thoroughly cleaned owing to lack of opportunity, the dirty boilers, the terrible fatigue due to the prolonged hard work, made the combustion of coal per square foot of grate surface comparatively low. In practice, it was found that the

maximum speed (with blowers in use) was not far from that obtained ordinarily on natural-draft trials."

Lieutenant Robison's graphic description refers to the fleet in southern waters throughout the war, during which period the ability to use full steam at short notice was required in order to be prepared, at first, for a chase and later, off Santiago, for the enemy's sortie. As the result, then, not of continuous high-speed steaming, but simply of "standing by" for it, with all, or the greater portion of the fires lit and full watches on duty, we find the speed and therefore the fighting value of the fleet reduced, 1st, by the fact that the boilers, being practically always in use, could neither be cleaned, scaled or repaired, with a consequent loss in efficiency; 2d, by boilers disabled from dropped crown sheets owing to salt feed as a "make up," since the evaporators were overworked in supplying continuously the full installation, leaking, as the latter was, from long service; 3d, by the severe and debilitating strain upon, and consequent loss of efficiency of, the fire room force when compelled to stand steaming watches for such a long period in intense heat; 4th, by the necessary expenditure of coal upon fires thus held in readiness, which expenditure probably cost the *Massachusetts* the loss of her opportunity to attack Cervera's squadron. It will be observed that this grave impairment of the fleet's efficiency was the result of the necessity, in "standing by," for keeping the fires lit in all or a large proportion of the furnaces, since a cold cylindrical boiler cannot be pressed into good steaming condition in less than three hours, although if, by the use of the hydrokineter, the water be previously heated to the boiling point, this time may be considerably reduced.

This inherent defect of the cylindrical boiler had a marked, although by good fortune not a disastrous, influence during the battle of Santiago. It was often impossible for the blockading vessels to coal at sea off that port, and retiring for that purpose to the shelter of Guantanamo Bay meant the absence of the ship for a time, and possibly from a battle line. It was not unnatural, therefore, that these conditions and that of the boilers themselves led to an unwise economy in the fleet's preparedness for full

steam. Thus, when Cervera's squadron appeared, the *New York* had "steam on four of the six boilers and hot water in the fifth; the sixth was ready for lighting fires." The captain of the *Brooklyn* "ordered steam on all boilers," and her speed rose gradually until, at the end of the four-hours' fight, she was making nearly 16 knots. The *Indiana*, during the ten days immediately preceding the battle, and probably also at the beginning of the latter, had "medium fires partly banked" in 26 of her 36 furnaces. The *Colon's* full-power trial speed was 19.35 knots. If her hull and machinery had been in good condition and the ship had been well handled, she could have outrun her pursuers and might have escaped. In any event, taking our vessels as a whole, it is an unquestionable fact that to the cylindrical boiler is chargeable a marked reduction of the speed of our fleet in battle, the crisis for which its ships were built. The single exception to this is the performance of the *Oregon*, which developed an engine power greater than that of her contract trial. She was not only handled with consummate skill, but fortune favored her. Unlike the rest of the drifting fleet, she had completed, but forty days before, a continuous run, stopping only for coal, of 14,500 miles at moderate speed—conditions most favorable for the preservation, with routine care, of boiler efficiency and for the thorough training of her fire-room force. Again, she had a reserve of selected coal stored in her "fighting bunkers," and, finally and vitally, the fires under her main boilers were, at the beginning of the battle, all "moderately heavy and spread all the time;" that is to say, the cylindrical boiler was given no chance to delay her in the chase.

THE MILITARY VALUE OF THE WATER-TUBE BOILER.

1. *Rapidity of Raising Steam.*—Steam can be raised in a well designed water-tube boiler, cold but with furnaces heavily primed, in from fifteen to twenty-five minutes after lighting fires. With this fact in view, it is clear that if, during the war with Spain, our ships had been fitted, like some of later date, with efficient boilers of this type, much loss of the fleet's speed and much suffering of the fire-room force would have been prevented. In

"standing by" for full speed at short notice, only part of the boilers need have been under steam; the rest, with furnaces primed would have been available for steaming with less than one-sixth of the delay required for cylindrical boilers under similar conditions. There would have been more or less time for overhauling the boilers not under steam, thus maintaining the ship's efficiency in speed, and full watches in the terrific heat of the fire-rooms would not at all times have been required. Admiral Melville's views as to this are:

"The fact that water-tube boilers will raise steam quickly is of the greatest advantage. I have stated elsewhere that I consider the battle of Santiago to have developed the necessity of the use of water-tube boilers, whether it taught us anything else or not. It would have been of the greatest advantage to have had, during the blockade of Santiago, boilers capable of raising steam in less than half an hour. Coal need not have been used to keep all the boilers under steam all the time. The *Massachusetts* might have shared in the glories of the fight if she had been fitted with water-tube boilers. The *Indiana* would have kept up with the *Oregon* and the *Texas*. The *New York* would have developed at least three knots more speed and the Navy would have been spared a controversy. I think the *Colon* would not have gotten so far away as she did. But we did not have the water-tube boilers."

Admiral FitzGerald, R. N., in the paper previously referred to, says:

"The *Terrible*, on January 9th, was lying in Plymouth Sound with her head to the eastward. At 8·25 she got under way and had to turn round to get out by the western channel; and at 9·08, that is to say in 43 minutes, she was going 20 knots. This performance could not have been approached had she been fitted with cylindrical boilers. * * * The *Sharpshooter*, on December 12, was lying at anchor, fires out, water in boilers cold. The fires were lighted, and in 14 minutes and 40 seconds the steam gauges showed a pressure of 60 pounds. The capstan engine was started, the cable hove short, and finally, in 35½ minutes from lighting fires, the ship proceeded at full speed."

There can be no question that the ability of the water-tube boiler to raise steam rapidly is of great tactical advantage. Owing to the small volume of contained water, this ability implies also that of making sudden reductions in speed without waste of steam.

2. *Endurance Under Strong Forced Draft.*—Closely allied, in tactical importance, with rapidity of raising steam, is the ability of a boiler to stand heavy forced draft. In fact, a boiler which cannot endure the latter is unfitted for naval installations. Forced draft is of value, not only in attaining maximum speed in emergency, but also in burning the inferior coals which naval vessels, in their widely extended service, are sometimes compelled to use. As a general rule, in powering cruising vessels, it should be treated as an aid in high-speed steaming only —"a reserve power which may be invoked in time of need," as the French Minister of Marine puts it—since, especially with the closed-stokehold system, the added and severe work of the fire-room complement, the rapid clogging of the grate bars and the loss of efficiency in firing and in evaporation per pound of fuel, unfit this powerful auxiliary for use in general naval steaming. However, for torpedo craft and for cruising vessels when pressed, it is essential. The battle of Santiago was fought under forced draft. The increase in power developed which it gives and its added cost in coal are shown, for cylindrical boilers of late design, by the following comparison of the performance of the battleship *Kearsarge* on her recent trans-Atlantic run under natural draft and that of her contract trial under an air pressure of 1.089 inches of water:

	Mean speed, knots.	I.H.P., total, all mach'y.	I.H.P., per sq. ft. G. S.	Coal, lbs. per I.H.P. per hour.	Coal per sq. ft. G. S., per hr.
Natural draft.....	13.1	5,375	7.85	2.17	17.
Forced draft.....	16.8	11,954	17.45	2.36	41.18

The data given for the natural-draft run are average results.

It will be seen that forced draft is thus vitally important to the warship in developing the maximum evaporative capacity which,

in a given boiler, the structure, circulation, and heat production and utilization permit. Experience has shown that its use, except under low limits as to air pressure and duration, is inadvisable with cylindrical boilers for structural reasons. Even under natural draft the effects of heat upon such a composite and relatively rigid structure are severe. The furnaces, unequally heated at top and bottom, not only bend but expand longitudinally, with consequent stress upon the seams at their ends; the tubes increase both in diameter and length, the latter producing thrust and movement at the tube-sheet joints; the seams of the shell suffer from both tangential and longitudinal stresses varying throughout; and the flat surfaces of the combustion chamber, tube sheets and boiler ends have complex strains and changes of shape. With forced draft and the consequent thick fires and absence of air for dilution, the temperature of the furnace is much higher and the heat effects are increased, the most serious results being the leakage at tube joints and furnace seams and the dropping of crown sheets.

In the early history of the closed-stokehold system, high rates of forced draft and of combustion per square foot of grate prevailed with cylindrical boilers, both of the return-tube and locomotive type, and with frequent disaster. In not a few instances British war vessels were unable to complete their four-hour, forced-draft trials. The boilers of the *Barham*, for example, leaked so seriously that she broke down and was towed home with hot water "over the stokehold plates and all of the stokers standing in ash buckets to keep their legs from being scalded." In the original *Charleston* of our Navy, whose boilers were of English design, the utmost limit of forced draft, under the closed stokehold system, was found to be ten hours, when "the tube-ends became so clogged with salt deposit that half of them were out of commission. The combustion chambers were also a foot deep with ashes and soot." The cylindrical boilers designed by the Bureau of Steam Engineering have had, owing to ample heating surface and low rates of forced draft, no such misfortunes as their British brethren; but, even as to them, Admiral Melville says: "As low a pressure as half an inch, will, after a day or two, begin to develop leaky tubes."

Water-tube boilers are, in elasticity of structure, greatly superior to the cylindrical type. There are neither furnaces nor shell forming an integral part of the boiler, and the tubes have, in the various designs, more or less freedom of expansion without endangering the joints. Neither this elasticity nor the areas of heating and grate surface form, however, the final measure of the maximum evaporative capacity of any boiler. That the latter may be forced with reasonable economy, there are required, in addition, a sufficient volume of furnace and combustion chamber for burning the gases, an effective system of baffling or tube-walls to lead the products of combustion over the entire heating surface, and, finally, a continuous and more or less vigorous circulation of the contained water that the heat may be conducted from the tubes with sufficient rapidity to prevent the formation of steam pockets or blisters with consequent burning of the metal. In the combination of these various elements in existing water-tube designs there is wide diversity. Therefore, for any given boiler and fuel it is evident that there is some particular rate of combustion at which boiler-efficiency is a maximum, and that, at other rates above or below, the efficiency will fall off. Forcing of practical value does not consist in a heavy expenditure per hour of coal upon the grate and of heat units at the top of the smoke pipe, but in an increased rate of evaporation from the heating surface and of combustion on the grate, without an excessive decrease in the number of pounds of water evaporated per pound of coal.

The cylindrical boiler is not only a rigid construction but the possible variation in the combination of the elements noted above is relatively small. The maximum furnace diameter is fixed by the limitation in plate thickness to prevent cracking and collapse; its length, and therefore that of the tubes, by the requirements of draft and efficient firing. The number of the furnaces depends upon the diameter of the shell and that, in turn, is regulated by the room available and, as a maximum, by the thickest shell sheet which it is possible to roll and rivet. The diameter and spacing of the tubes are, in naval practice, standard, and their number is governed by the needs of steam room and water

circulation. In the water-tube boiler, on the other hand, there is, owing to the omission of the shell and the separation of the furnace and tubes, wide latitude in practically all of the elements of design, excepting the length of the furnace, its width in some makes, the tube-inclination, and the maximum ratio of tube length and diameter. As a consequence, in addition to the possible elasticity of construction, there is the further possibility of so proportioning furnace volume and heating surface and so adjusting baffle plates and circulation as to permit a high degree of forcing without structural detriment. The water-tube boiler is, therefore, essentially a forcing type, although, in some forms this quality exists in but moderate degree. For the varying powers required in general naval service, a boiler should have not only high efficiency at a medium rate of combustion but should maintain that efficiency, without serious loss, at a fairly high rate of coal consumption. The practice at present is to restrict strong forced draft to torpedo craft, and to use in cruising vessels an air pressure not exceeding, as a rule, one inch of water.

Type.	Authority.	Rate of combustion of coal per sq. ft. grate, lbs. per hour.	Evaporation f/a 212° per lb. coal.	Evaporation reduced to standard rate of 30 lbs. coal consumption.
Belleville	British Boiler Committee	15.8	11.46	10.85
		19.8	11.03	10.61
	Bertin	46.	8.4	8.9
Niclausse	British Admiralty test	12.8	10.15	9.5
		22.	9.6	9.31
	Bertin	15.	8.9	8.4
		46.	8.4	8.9
Thornycroft	Catalogue	7.74	11.22	10.29
		66.8	8.89	10.1
Lagrafel D'Allest	Milton	12.3	10.068	9.4
		45.76	8.24	8.72
Babcock and Wilcox	U. S. N. test	19.61	11.03	10.60
		59.23	9.63	10.68
Guyot-Du Temple	Bertin	22.	9.4	9.12
		46.	8.4	8.9
Hohenstein	U. S. N. test	18.3	10.16	9.716
		71.7	8.29	9.68

The table on the preceding page, for which the writer is indebted to Mr. Alfred H. Raynal, gives the efficiencies of various water-tube boilers at different rates of combustion.

3. *Rapidity of Construction and Repair.*—It is difficult to measure the vast responsibility of the United States, not only in guarding an extended coast line and distant insular possessions, but in the maintenance of the Monroe doctrine with regard to the American continent and its adjacent islands. With interests so varied and of such magnitude, wars for their protection are inevitable; and, in meeting European, and possibly Asian enemies, these wars will be largely naval—a struggle, in the beginning at least, for the command of the sea. It may be safely said as well that no contest of the future will be won with the ease of that with Spain, a gallant but feeble and most incapable foe.

Looking backward, it is possible to realize in some degree the hurried purchase and construction which, in facing war with a strong naval power, may prevail. At the beginning of the Civil War there were available but 69 vessels, 34 of which were sailing ships. By the end of the year 1861, there were afloat 211 vessels carrying 2,301 guns and 20,000 men. During that year the construction of 52 steamers was begun, including 23 "ninety-day gunboats," whose popular name shows the stress and haste of that time. At the close of the four years' war, the force afloat consisted of about 600 vessels of all classes.

While our Navy will probably never again be so disorganized and unprepared, one of the main lessons from this record is the imperative necessity of speed in the emergency construction and repair of war vessels. The water-tube boiler, owing to the sectional structure of some types, is especially adapted to meet this requirement. Speed construction in the renewing of boilers is a military factor of importance that is now commanding the attention of naval strategists. In his report for the year 1898, the then Engineer-in-Chief said:

"One very striking advantage for war vessels is that the (water-tube) boilers can be replaced or practically rebuilt without disturbing the decks, all parts being small enough to pass through

permanent openings. This was the case in the *Monterey*, as mentioned last year, and it was strikingly illustrated this summer in the case of the *Canonicus*, *Manhattan* and *Mahopac*, when it would have been impossible to use any but water-tube boilers without practically rebuilding portions of the hull."

The Ward boilers of the *Monterey*, to which Admiral Melville refers, were retubed in place by the ship's force. The three single-turretted monitors had rectangular boilers, for which were substituted those of the Babcock & Wilcox type, the aggregate power being 4,500 I.H.P. The speed in construction made possible by the water-tube system is shown by the fact that two of the vessels were ready for steam in 30 days and the third in 42 days after the order to proceed with the work was received. The contrast between this performance and that with the cylindrical boilers of the battleship *Indiana* is marked. It is stated that four months were spent in renewing the furnaces of her boilers.

As a further instance of the rapidity with which water-tube boilers may be built, it may be said that the makers of a prominent boiler of this type replied to an official inquiry that they could furnish to the U. S. Navy in thirteen months boilers to develop 82,000 H.P. without interference with other large naval installations then building.

The possibility of ready and rapid repairing may be regarded as a frequent characteristic of water-tube boilers. Owing, however, to the great variation in design, there are marked differences in this respect between the several types. Each of the latter must be judged as to this by the details of its construction and the effect of service upon its parts.

4. *Weight Saving.*—The lightness of the water-tube boiler and its contained water, as compared with the cylindrical type, was the characteristic which, primarily, forced its introduction in warships, since any lessening in weight of the boiler installation may be utilized to increase the speed, the radius of action, the armament or the armor of the vessel. Thus, the "Memorandum respecting Water-Tube Boilers in H. M. Ships," presented in 1900 to the British Houses of Parliament, states:

"If the Admiralty of the day had not decided to put water-tube boilers into the *Powerful* when she was designed, or the subsequent successive, Boards had hesitated to follow their policy, and had waited until the whole of the minor difficulties involved in the change had been overcome, the magnificent fleet which has been built, building and projected since that time, consisting of 20 battleships, 22 armored cruisers, 10 protected first-class cruisers and 9 second-class cruisers, making a total of 61 ships, would have had at least a knot less speed than they will now have, or an equivalent sacrifice would have had to be made in other directions, which, to gain a knot at the high speeds now necessary, would be very considerable in amount."

The improvement in operation and performance of the Belleville boiler, which, despite the complication and inherent defects of the latter, have been attained in the British navy, give this statement substantial support. While this boiler has marked advantages in weight saving, it is a marvel of complexity and of ingenuity in the various mechanisms applied because of an almost stagnant circulation, a furnace of insufficient volume, the absence of steam room, the production of wet steam and the necessity for automatic feeding. Yet, when handled with skill by a trained force, it is, at moderate rates of combustion, both safe and successful, and is, in military value, distinctly superior to the cylindrical type. However, in other forms of the water-tube boiler, steam is generated efficiently without undue complexity and with an important saving in weight.

This economy in weight lies in both that of the boiler and that of its contained water. Thus, to take extreme cases, the battleship *Wisconsin*, 12,609 I.H.P., with eight single-ended cylindrical boilers, has a total weight of boilers and water of 96.64 pounds per I.H.P., of which total 27.2 per cent. is water. On the other hand, the torpedo boat *Thornton*, 3,000 designed I.H.P., with three Thornycroft boilers, has a total weight of boilers and water of but 29.21 pounds per designed I.H.P., of which total 13.88 per cent. only is water. These relative weights are shown, for cruising vessels, by the following table:

Ship.	Boilers.	Steam pressure.	Weight in pounds, boilers and water.		
			Per sq. ft. G.S.	Per sq. ft. H.S.	Total per I.H.P.
B.S. *Oregon*.....	Cyl. 4 D. E.....	160	1,690.5	53.44 }	94.68
	Cyl. 2 S. E.....	160	1,853.5	61.22 }	
B.S. *Wisconsin*.	Cyl. 8 S. E.....	180	1,904	45.98	96.64
B.S. *Maine*.....	24 Niclausse.....	250	1,045.3	24.34	89.28
B.S. *Missouri*..	12 Thornycroft..	250	752.5	14.23	44.93
B.S. *Nebraska*..	12 B. & W.....	265	1,018.4	24.32	71.93
M. *Florida*.....	4 Mosher.....	250	525.6	13.27	52.67

Again, the limit of working pressure of the cylindrical boiler is about 210 pounds. Beyond that pressure the shell sheets become too thick to be worked with safety, the furnaces become too small in diameter and the boiler is excessively heavy. Since the advantage of an advanced steam pressure cannot, within reasonable limits, be disregarded, the only just comparison between the cylindrical and water-tube types is that at the working pressure which the latter makes practicable. The following data show this relation with regard to a naval installation of 16,500 I.H.P. and 250 pounds steam pressure.

	6 D. E. and 2 S. E. Scotch boilers. 250 pounds.	12 B. & W. boilers. 250 pounds.
Floor space, fire-room, total, square feet.....	5,207	4,968
per I.H.P., sq. ft.....	0.3156	0.311
engine-room, square feet..	2,024	2,024
total, square feet.....	7,231	6,992
Weight, boilers and fittings, tons.....	739	462
water in boilers, tons.....	252	105
boilers and water, tons	991	567
all machinery, including stores, tons..	2,045	1,630

It will be seen that the use of Scotch boilers would, as compared with the water-tube type cited, increase the total floor space by 3.4 per cent., the total boiler weights by 424 tons, or 74.8 per cent., and the total machinery weights by 415 tons, or 25.5 per cent. Furthermore, the Scotch boiler at this pressure is, at present, practically an impossibility at any reasonable size. Thus, the data of the table refer to boilers 15 feet $5\frac{1}{8}$ inches diameter,

whose shell sheets are $1\frac{59}{64}$ inches thick, braces 3 inches in diameter, and furnaces but 2 feet 9 inches diameter.

In considering the relative weights of the various types of water-tube boilers, there should be borne in mind the fact that the uniform success of U. S. Naval cylindrical boilers was owing to their ample strength for the work required of them, which strength necessitated more weight per I.H.P. than that allowed in some other navies. Therefore, in securing the more important advantages of the water-tube system in other respects, the gain in weight saving should not be made excessive by light casings, which will warp and leak; by thin tubes, which will corrode rapidly; and by too small a volume of contained water, which will make the boiler sensitive in feeding, operation and delivery of steam.

5. *Less Danger from Explosion.*—The water-tube boiler, as compared with the cylindrical type, gives a decreased danger from explosion if struck by shell during an engagement. The benefit from this characteristic will, however, be infrequent, since the boilers are shielded by the thickest side armor and by the protective deck. In any event, the danger from the explosion of any boiler lies in the volume of contained water heated to the temperature corresponding with the steam pressure. In the cylindrical boiler this volume is large, and an explosion would injure most seriously any vessel, while, in the water-tube type, the volume is not only much smaller per I.H.P., but the number of boiler units is frequently greater and the size of each less for the same total power. It is true that, in recent years, serious and sometimes fatal accidents have occurred with the Belleville boiler in the British navy and with the Niclausse boiler in American, French and Russian vessels. These have been due to the bursting of tubes, however, and the disaster has been confined, in each case, to the fire room affected.

OTHER ADVANTAGES OF THE WATER-TUBE BOILER.

In addition to its tactical advantages, the water-tube boiler is, in comparison with the cylindrical type, superior in other important respects which may be noted briefly:

1. *General Flexibility of the Installation.*—While the height, for each type of water-tube boiler admits little variation, the furnaces, having greater freedom as to draft than one of circular section, may be made longer—50 per cent. in some cases—than the standard for Scotch boilers. Again, the width of water-tube boilers, whose tubes lie lengthwise of the furnace, is variable within wide limits, thus making possible any desired number of units. As to this, it may be noted that the Boiler Committee appointed by the British Admiralty advised in 1892 that the double-ended cylindrical boiler be abandoned in order to secure a greater subdivision of the power. Again, boilers whose tube inclination is sufficient to prevent the rolling of the ship from affecting the circulation, can be set either fore-and-aft or athwartship. This does not hold with the Belleville and Niclausse boilers, whose tubes must lie fore-and-aft. Finally, water-tube boilers whose transverse section is approximately rectangular are better adapted to fit the space available than the cylindrical boiler is, and there is, therefore, the possibility of installing a greater area of heating surface per square foot of grate.

2. *Ease of Cleaning Water Surfaces.*—In the cylindrical boiler the exterior tube surfaces are, in large degree, inaccessible, and when pitting occurs or scale is deposited the greater part of the tubes must be removed before the remainder can be examined or cleaned. This disadvantage does not, in general, exist with the water-tube type, although the degree of freedom of access varies considerably in the different forms. Thus, in the Babcock & Wilcox boiler, hand holes in the headers make the interior of the tubes as open to examination and cleaning as that of a cylindrical boiler. In the Belleville elements, doors in the front junction boxes give access. In the Niclausse boiler, both the generating and circulating tubes may be readily removed when the former tube is not bent. The Yarrow tubes may be inspected internally from the water box or steam chest by means of a small electric light passed through each tube. The bent tubes of the Thornycroft boiler permit only a partial and unsatisfactory examination by running a steel wire brush through each tube.

3. *The Practicability of Higher Steam Pressures.*—As has been

stated, the increase of steam pressure in the cylindrical boiler is, for the present at least, barred by the thickness of shell sheets, the required diameter of furnaces and the existence of flat, stayed surfaces. With the water-tube boiler there is no such limit, since the principal parts are cylinders of small diameter with joints well guarded, as a rule, from heat action. The increase in pressure thus possible gives not only higher steam economy, but smaller and hence safer piping, less diameter of some cylinders and a shorter engine.

OBJECTIONS URGED AGAINST THE WATER-TUBE BOILER.

The principal objections advanced against the water-tube boiler are:

1. *Lack of Economy in Continuous Service.*—The efficiency of the water-tube boiler in cruising war vessels under the condition of prolonged service is a question as to which general data is lacking, since extended experience with the type has not been had, excepting with the Belleville boiler abroad and, in small vessels, with the Babcock & Wilcox boiler in the U. S. Navy. The results of full-power contract trials indicate that, when in good condition and skilfully handled, the water-tube boiler is superior to the cylindrical type. Thus, the most favorable performance shown by the latter in recent contract trials in the U. S. Navy is 1.91 pounds of coal per I.H.P. per hour, while the best results during similar trials of the water-tube type are: Belleville, 1.55 pounds; Babcock & Wilcox, 1.57 pounds; Dürr, 1.56 pounds; Niclausse, 1.586 pounds.

Sustained efficiency, under the exigencies of active service is, however, another question, since it depends not only upon the boiler's qualities, but, as to its condition and operation, upon the skill of a limited fire-room force. During the British maneuvers of 1899 there were attached to the "A" fleet nineteen vessels having cylindrical boilers and five fitted with the Belleville type, four of the latter ships having been for some time in commission. A three-hour trial of this fleet, at three-fifths N.D. power, gave average results, as to the boilers, in pounds of coal per I.H.P. per hour, as follows: Cylindrical, 2.27 pounds; Belleville, 2.23

pounds. Again, during the recent run of the British cruisers *Spartiate* and *Europa* from Portsmouth to Hongkong and return, a distance of about 20,000 miles, the average coal consumption of the former vessel was 1.83 pounds per I.H.P. per hour and of the latter 1.91 pounds. Both ships are fitted with Belleville boilers, a type requiring unusual skill in management.

In the U. S. Navy the only comparison possible with similar ships is that of the five gunboats given below, which are of 1,000 tons' displacement, and were launched about seven years ago. Admiral Melville gives the following table, showing the relative economy of the cylindrical and water-tube boilers in these ships on cruising duty :

Ship.	Boilers.	Knots per ton of coal at most economical speed.
Annapolis	2 Babcock & Wilcox	26.38
Marietta	2 Babcock & Wilcox	22.27
Newport	2 S. E. cylindrical	18
Vicksburg	2 S. E. cylindrical	21.25
Wheeling	2 S. E. cylindrical	16.6

2. *Necessity for Fresh Feed Water.*—It has been urged against the water-tube boiler that the necessity for pure feed water and the consequent weight of the evaporators required, nullified the gain in weight-saving by the use of the water-tube system. In degree only this is true. Modern practice makes fresh feed essential for cylindrical boilers also. In his report for the year 1898, the then Engineer-in-Chief, said : " Fresh water for the boilers is almost as important as coal ;" and, in a later paper : " Up to this time we have had no trouble from salt water or grease in water-tube boilers. Indeed, we could hardly be more troubled by salt water with this type of boiler than we have been with cylindrical boilers. We suffered severely, in our short war with Spain, from dropped furnaces in cylindrical boilers." It may be noted also that the battleship *Kearsarge*, in her recent trans-Atlantic run, carried 55,000 gallons of fresh water in the double-bottom compartments under the fire rooms, as " make-up " feed for her cylindrical boilers.

It is true that, in boilers of the water-tube type, owing to the small volume of contained water and the use of small tubes in some designs, the effects of impure feed are felt more quickly than in cylindrical boilers. Water tubes are subjected to intense heat, and an assured circulation is necessary to prevent the ultimate burning of the metal. On the other hand, however, small tubes are usually so placed as to have a circulation so vigorous as to act as a scour, and a number of instances have shown that salt feed may be used, temporarily at least, in some boilers having large tubes, without disaster. Thus, Messrs. Yarrow & Co. tested one of their torpedo-boat boilers having 1-inch tubes with salt feed for a period of 7 to 8 hours on each of five consecutive days. The boiler worked satisfactorily throughout, the thickest scale deposit being $\frac{1}{32}$ inch on the lower half of the drum and inside of the tubes. The boiler was cleaned several times during the trials. It is said also that the Thornycroft boilers of the torpedo boat *Ericsson* were repeatedly fed with salt water while off Santiago and yet escaped injury. Ensign Dringer states, as to the Babcock & Wilcox boilers, of the *Marrietta*, that "Salt water sufficient to bring the saturation up to $\frac{1}{32}$ has been used. The ship has cruised at full speed many times for several days with this saturation without any bad results at the time or afterwards. However, every effort was made to maintain a fresh feed at all times." Even the Belleville boiler, with its restricted circulation, will stand salt feed for a time, if blown off frequently. The Russian armored cruiser *Minin*, on her first voyage with these boilers, went from Kronstadt to the Piraeus without evaporators on reserve feed, and the boilers were fed with sea water. Upon arrival in port, the tubes were examined, and only in the two lower rows was there found scale of $\frac{1}{32}$ inch in thickness.

While the unnecessary use of salt feed in any boiler is universally condemned by engineers, it is essential that war vessels shall be equipped with boilers which, in emergency, may be fed with river or sea water. This quality, although not common to all of the water-tube type, may be secured by proper design and construction.

3. *The Disablement of the Boiler by the Bursting of a Single*

Tube.—The water-tube boiler is distinctly inferior to the cylindrical type in that the splitting of a single tube will put the boiler out of service until the fires are hauled, the water blown out and the tube is plugged or replaced. With this defect, however, there should be considered the readiness, in certain types, with which repairs can be made and the rapidity with which steam can be again raised. Thus, in the Babcock & Wilcox boiler, tubes can be plugged with the same ease as in the cylindrical type. During British tests of these boilers on the torpedo gunboat *Sheldrake* several hand-hole caps were removed in 24 minutes after the fires were hauled and the water blown out from a boiler which had been working at full power. Three tubes were then drawn in an average time of 10 minutes each. As to the Belleville boiler, M. Bertin states that the replacing of an element in which a tube has given way can be accomplished in two hours, and, in the absence of spare elements, the damaged tube may be replaced in four or five hours. In the Du Temple boiler, when the tube ends are, as formerly, connected with the drums by two nuts, the tube is closed by substituting a blind nut for the inner of the two, the operation, owing to the necessary cooling of the boiler, taking four hours from the time the fires are hauled until the boiler is refilled. If the threads are in bad order the tube must be cut and plugged, which requires at least an hour more. A tube was replaced in a Niclausse boiler, previously under steam, in 25 minutes during British naval tests. When, however, the generating tubes of this type are bent from overheating the difficulty of removal is considerable.

CONCLUSION.

It seems clear the water-tube boiler, in some forms at least, has reached a stage of development which makes the use of the cylindrical type in war vessels a grave military error in the sacrifice of such advantages as rapidity in raising steam, forcing in emergency without detriment, swift construction and repair, and weight-saving which can be utilized in added speed, armor or armament. The cylindrical boiler has so blocked progress in these paths that its usefulness as a war instrument seems

ended. As to the objections urged so often against the water-tube type with regard to complexity and sensitiveness in operation, it may be said that these conditions are relative only and vary widely with different boilers. Even if this were not so, such objections cannot be considered as a bar to added military strength. Nelson's short and sightless smoothbores, with their excessive windage to allow for hot shot, were somewhat less complex and less difficult of operation than the massive rifles of to-day with their actuating mechanism. The leading boiler need of the present seems to be not so much further development as the thorough training of the fire-room force to meet new conditions and to maintain the trial efficiency of the water-tube boiler in cruising duty.

REPORT OF BOARD OF U. S. NAVAL OFFICERS ON BOILERS OF U. S. S. *MAINE*.

NAVY YARD, NEW YORK, N. Y., *October 29, 1903*.

SIR:—In obedience to the Department's order No. 1958–5, of September 30, 1903, the Board to witness and report upon the performance of the *Maine's* boilers during the voyage of that vessel from Newport News, Virginia, to Culebra and return, met on the U. S. S. *Maine*, at Newport News, Va., Monday, October 12, 1903.

1. The instructions to the President of the Board from the Secretary of the Navy were read and discussed. Additional instructions from the Secretary of the Navy to the Commanding Officer of the *Maine* were referred to the Board by that officer for its information.

2. The Board made a preliminary inspection of the boilers to note their general arrangement, and to see the location of the water gauges and automatic feed devices. The most convenient place to make an opening for the pyrometer was found to be at a point about 30 feet above the furnace grates in each smoke pipe.

3. *Boilers.*—The *Maine* has twenty-four Niclausse boilers, arranged in three groups of eight boilers each, with one smoke pipe for each group. Each boiler is composed of fifteen elements, each element containing twenty-four generating tubes, $3\frac{1}{4}$ inches outside diameter by 7 feet $3\frac{3}{16}$ inches long. The generating tubes are No. 8 gauge in thickness, and are secured into a malleable-iron lantern at one end and closed at the other end by a threaded cap. Inside each generating tube there is a circulating tube $1\frac{9}{32}$ inches outside diameter, and No. 21 gauge thick.

4. The headers, of malleable iron, are secured to a cylindrical drum extending across the front of the boiler.

5. The drums are 2 feet $7\frac{1}{32}$ inches inside diameter, and are 9 feet $2\frac{5}{32}$ inches long.

6. *Smoke pipes.*—There are three smoke pipes, each 9 feet 10 inches in diameter and 100 feet high, measured from the grate.

The acceleration of draft due to these high smoke pipes is about .3 inch of water.

7. The total grate surface of the twenty-four boilers is 1,353.6 square feet; the total heating surface is 53,343 square feet.

8. There is one furnace to each boiler, with three furnace doors. Each furnace is 8 feet 4 inches wide and 6 feet 6 inches deep.

9. The boilers were designed to furnish steam for 16,000 I.H.P. of main engines, air and circulating pumps, and an additional I.H.P. for the usual auxiliaries fitted on a battleship. This total I.H.P., the Board is informed by Mr. Sneddon, of the builder's firm, was figured at 17,848 I.H.P., on the basis of the water consumption of the main engine, assumed at 17 pounds of water per I.H.P. per hour, at full speed. The designed steam pressure is 250 pounds in the boilers and 200 pounds at the main engines.

10. *Forced Draft.*—There are twelve Sturtevant blowers for forced draft on the closed fire-room system. These blowers have never been used for forced draft since the ship has been in commission. On this voyage of the *Maine* these blowers were not used for any purpose. They cannot be used for ventilating the fire rooms, because the hot air rising from the fire-room hatches is immediately taken through the blowers and blown back into the fire rooms. If the ventilators located inside the superstructure should be raised high enough to clear the boats stowed on the cradles, the natural ventilation of the fire rooms would be improved.

11. The usual auxiliaries in a vessel of this class were in operation during the voyage.

12. The Board spent a portion of its time in the fire rooms while the vessel was under way, and also while in port, and noted the performance of the boilers, as well as the skill and attention of the fire-room force.

13. On Tuesday, October 13, 1903, about 11 A. M., the *Maine* left Newport News, Virginia, and proceeded to sea, bound for Culebra. All the boilers were in use, and the engines were gradually worked up to 110 revolutions per minute, increased the next morning to 115. One representative of the Stirling Boiler Company, who built the boilers, and three representatives

from the works of William Cramp & Sons, were on board for the voyage. At 4·39 P. M., October 13, 1903, the run to Culebra was held to have begun.

14. At noon, October 13, the coal on board, according to the steam log, was $1{,}512\frac{1960}{2240}$ tons. The coal was Pocahontas of good quality.

15. The D. B. compartments, including "B" 99, called the reserve-feed tank, contained about 80,000 gallons of fresh water. Sixty thousand gallons of this water were taken on at Newport News from a water boat, after being tested for salt.

16. In addition to the regular engineer's force of 225 men, 78 firemen, coal passers, oilers and machinists 2d class, were taken on temporarily for the run. The additional oilers were experienced men from the U. S. S. *Texas* and the torpedo boats in reserve at Norfolk, Va. The following table shows the complement allowed for the various ratings, and the numbers in those ratings available for duty October 13, 1903:

	Complement.	*Available.*
Chief yeoman,	1	1
Chief machinists,	6	6
Machinists, first class,	5	5
Machinists, second class,	5	9
Boiler makers,	2	2
Blacksmith,	1	1
Coppersmith,	1	1
Water tenders,	18	18
Oilers,	14	25
Firemen, first class,	38	39
Firemen, second class,	49	68
Coal passers,	85	128
Total,	225	303

The engineer officers are one lieutenant commander and two lieutenants. There are also four warrant machinists.

17. The voyage divided itself into two parts: the run down, 78.37 hours, at an average speed of 15.1 knots, and the run back, 76.3 hours, at an average speed of 15.9 knots. The run down is

further divided into two parts—one with all the boilers (24) for a period of fifty-two hours, at an average speed of 16.64 knots, and the other with fourteen boilers for a period of 26.37 hours, at an average speed of 12 knots. This slowing down was not the fault of the boilers, but was caused by trouble with the main engines, as stated below.

18. At 8·39 P. M. October 15 the starboard engine had to be stopped on account of increasing pound in H.P. and I.P. crank-pin brasses. Started up again at 9·31 P. M., but at reduced speed and with fourteen boilers in use. The reduced speed was made necessary by the condition of the crank-pin brasses of the starboard engine. This reduced speed was maintained until 1 A. M. October 17, when the run to Culebra was held to have ended.

19. According to the steam log, the coal used for all purposes for the fifty-two hours of the run south, with all boilers, was $544\frac{2190}{2240}$ tons; for the 26.37 hours of the run with fourteen boilers, the log account of coal used was $169\frac{1840}{2240}$ tons. Total, $714\frac{1790}{2240}$ tons.

20. Careful inspection of the bunkers upon arrival at Culebra showed that the coal account was 65 tons short. Adding 65 tons to the coal account given above, in proportion to the hours of the two runs, gives for the fifty-two hours with all boilers, 1,353.6 square feet of grate surface, $588\frac{241}{2240}$ tons used for all purposes, and for the 26.37 hours with fourteen boilers, 789.6 square feet of grate surface, $191\frac{1549}{2240}$ tons. Total coal for the entire run south, $779\frac{1790}{2240}$ tons.

21. The corresponding rates of combustion per square foot of grate are, respectively, 18.72 pounds and 20.62 pounds.

22. The smoke-pipe temperatures for the fifty-two hours when all boilers were in use averaged 442 degrees Fahrenheit for the forward pipe, 389 degrees Fahrenheit for the middle pipe, and 472 degrees Fahrenheit for the after pipe. These temperatures were taken at a point thirty feet above the grates. The smoke-pipe temperatures were not taken for the run with fourteen boilers.

23. The ashes and refuse from the coal for the entire run south was $97\frac{1820}{2240}$ tons.

24. Two sets of indicator cards from the main engines were taken daily during the run. The average indicated horsepower of the main engines during the fifty-two hours was 10,188. The I.H.P. of the auxiliaries in use has been estimated from the trial trip data as 750. The coal per I.H.P. of main engines and auxiliaries was 2.31 pounds.

25. Two evaporators, of a rated capacity of 9,500 gallons each, for twenty-four hours, were kept on the make-up all the time, and they were equal to making good the losses of fresh water. One and sometimes two additional evaporators were kept on the ship's tanks, distilling about 4,000 gallons per day. Remarks on the efficiency of the evaporator plant follow later in this report.

26. The boilers on the run south gave no trouble at all. The fires were carried as evenly as possible and moderately heavy, in accordance with the Department's instructions. No attempt was made to force the fires. There was difficulty at times in keeping up the steam pressure for 115 revolutions of the main engines, and the engines fell below the required revolutions at times. It should be noted, however, that the cut-offs of the main engines were not set for economical running, nor were the feed-water heaters in use.

27. Since the coal burned per square foot of grate for the fifty-two hours with all boilers in use was only 18.72 pounds, the principal trouble in keeping steam must have been the inexperience of the firemen. The draft was good and the fire rooms were not oppressively hot. It was evident that many of the firemen were without experience, and certainly the low rate of combustion attained could not have overworked anyone. Two boilers of this ship, running the usual auxiliaries used in port, have often been worked to a higher rate of combustion than was shown on the run south.

28. There did not appear to be any system for regular firing, the firemen throwing on coal at such times and in such quantities as seemed necessary to each individual. When the steam pressure would be low the firing was done as it would be done

with Scotch boilers, not as it should be done with water-tube boilers. Some of the boilers appeared to be doing more than their share of the work. This was particularly true of the boilers in the forward and after fire rooms, where the draft was strongest. The firemen were young and strong, but lacked training with water-tube boilers.

29. The need of a responsible head to direct operations in the fire rooms was plainly apparent. There should be a chief water tender for each fire-room watch, in addition to the regular water tenders, with a well-defined system of firing and with the ability to see that this system is rigorously carried out. The multiplicity of units apparently necessary with water-tube boilers makes systematic firing absolutely essential. The old system of firing will not accomplish the necessary result of making each unit do its proper share of the work. It is necessary for the Navy to train firemen for water-tube boilers, because so few of these boilers are used in the merchant service that our supply of trained men from that source will always be limited.

30. At Culebra repairs to the main engines were taken in hand as soon as steam was off the main steam pipes. The necessity for taking on coal and repairing the main engines and evaporators kept the ship at Culebra until Tuesday, October 20. Advantage was taken of this stay in port to make an examination of the boiler tubes.

So far as could be seen all tubes were straight except one in boiler "B," slightly bent, and one in boiler "E," bent up about $\frac{3}{4}$ inch. A new tube was put in boiler "E." The bent tube was opened and inspected. It was found to be perfectly clean. Boilers "A" and "B" had been fed with a slight quantity of salt water several days before leaving Newport News. The saturation in these boilers had shown one-half to seven-eighths, so that it was thought best while at Culebra to look at the two lower rows of tubes. Most of these were found to contain scale from $\frac{1}{32}$ inch to $\frac{1}{16}$ of an inch thick, that easily fell out when the tube was up-ended and jumped upon the floor plate. The circulating tubes showed slight signs of grease. To save time, sixty new generating tubes were put into boilers "A" and "B," but the ones

taken out can soon be made ready for use by running the rotary cleaner through them. Most of these sixty generating tubes after being taken out were seen to be slightly bent, although this bending was not apparent until a straight-edge was applied. The time required to remove these sixty tubes, after the connection doors were down, and replace them with new ones, was six hours and twenty minutes. Fortunately, no tube turned, so there was no delay caused by having to put on a special tool for holding a tube that turns when attempt is made to unscrew the plug. One water tender and three helpers on each boiler, under the direction of a warrant machinist, did the work.

31. The absence of soot or dirt on the generating tubes of all boilers was noticeable. This is partly due to the strong draft, and, probably, partly to the vibration of the tubes, firmly supported as they are at one end only.

32. During the stay of the ship at Culebra the make-up feed required for the boilers in use as auxiliaries was taken from the water bought at Newport News, Va. This had to be done on account of repair work on the ship's evaporators. In some way, not clearly explained, salt water got into the feed water for boilers "X" and "Y," in use as auxiliaries. The saturation soon showed as high as $1\frac{1}{4}$. Fires were immediately started in other boilers, and, as soon as possible, boilers "X" and "Y" were shut off and the water freshened by pumping, and blowing with the bottom blow. This operation is necessary before opening any one of these boilers that has salty or dirty water in it, if it is thought necessary to get rid of the salty or dirty water, owing to the impossibility of emptying the boiler except by opening every tube.

33. As soon as these two boilers, "X" and "Y," were shut off, steam was allowed to drop, and then the pumping and blowing was proceeded with until the saturation showed $\frac{1}{4}$, or nearly fresh. This operation required about three hours. Fires were then hauled, the front doors taken down and the dirt brushed from the headers. Six tubes in the two bottom rows of each boiler were then opened, circulating tubes withdrawn, and the water blown out of the generating tubes by compressed air working through a hose thrust to the end of the tube. Very little scale

showing in these tubes, and all the tubes appearing to be straight, it was decided not to take out any tubes. The opened tubes were closed and the two boilers prepared for steaming. The time taken to get these two boilers in condition for steaming, with fresh water in them, was about nine hours, including the three hours of pumping and blowing. As there was no hurry, the operation was leisurely performed.

34. On Monday, October 19, coaling ship was in hand all day. The Board had hoped to get the coal used by the two boilers furnishing steam for all auxiliaries in use, as it was apparent that these two boilers were being worked hard to furnish the steam required. Owing to the salting, early in the day, of the water in boilers "X" and "Y," they were put out of commission, so that the coal account for these boilers could not be had with sufficient accuracy, or for a sufficient length of time, to give a correct idea of the rate of combustion maintained.

35. During the forenoon of Tuesday, October 20, 1903, the *Maine* left Culebra on the return run to Hampton Roads. All boilers were in use, but the engines were not speeded up for some time after getting under way. Departure was taken at 1·38 P. M., October 20, and the return run held to have begun at that hour.

36. The log account at noon, October 20, showed $1{,}299\frac{120}{2240}$ tons of coal in the bunkers; coal was Pocahontas of good quality.

37. The D. B. compartments, including "B" 99, called the reserve feed tank, contained about forty-six thousand gallons of fresh water. During the three days' stay of the ship at Culebra about 34,000 gallons of water were used from the D. B. compartments for make-up feed and for freshening boilers "A," "B," "X" and "Y."

38. At 10 P. M., October 20, the revolutions were increased to 100 per minute. Three water tenders were detailed to act as chief water tenders. Their duty was to make the rounds of the fire room to try to enforce a uniform system of firing. The result proved of great advantage, as the firing was much better done on the return run.

39. The revolutions of the main engines were gradually increased by increments of five until 110 revolutions were ordered at 5 P. M., October 21. This rate was easily maintained until 7 A. M., October 23, when 115 revolutions per minute were ordered. At 9·38 A. M., October 23, the revolutions were increased to 120 per minute. The boilers responded quickly to the varying demands for steam. At 11·13 A. M., October 23, the engines were slowed to 110 revolutions on account of cracks showing in "Y" frame of starboard L.P. cylinder. At 6·10 P. M., Currituck light being abeam, the run north was held to have ended.

40. There appeared to be difficulty in supplying the necessary make-up feed on the run north. A greater part of the time three evaporators were on the main condensers for make up, although no great amount of leakage was to be seen about the machinery. It is possible that the evaporators had begun to scale, and it was also noticeable that the evaporator men did not run the evaporators to their full capacity, as they were afraid of foaming.

41. According to the steam log the coal used for all purposes for the run north was $758\frac{1650}{2240}$ tons. Careful inspection of the bunkers on arrival at Hampton Roads showed that the coal account was correct. The total ashes and refuse for the run was $99\frac{440}{2240}$ tons.

42. The average rate of combustion per square foot for the run north was 16.45 pounds, and the average coal per I.H.P. figures 2.45. This poor result is due to the fact that through nearly all of the run north too many boilers were in use for the revolutions at which the main engines were run. For eight hours of the run, with revolutions 110, 115 and up to 120 per minute, the rate of combustion per square foot of grate was 21.98 pounds.

43. The average smoke-pipe temperatures were 414 degrees Fahrenheit for the forward pipe, 423 degrees Fahrenheit for the middle pipe and 433 degrees Fahrenheit for the after pipe. The highest temperatures were 570 degrees, 450 degrees and 510 degrees for the forward, middle and after smoke pipes respectively.

44. The boilers gave no trouble at all; they easily furnished steam for such speeds as the engines could maintain. Upon

arrival at Newport News, and after fires had been hauled, the boilers were inspected by the Board. Several generating tubes were taken out of every boiler and examined for straightness and for evidences of grease or scale. Most of the generating tubes taken from the two lower rows are bent from $\frac{1}{8}$ inch to $\frac{1}{4}$ inch. These bent tubes were clean. Several circulating tubes were found to contain considerable greasy sludge. The circulating tubes in which the greasy sludge was found were, in nearly every case, those near the bottom blow outlet. So far as could be seen, the boilers were in good condition after the voyage, and ready for another run.

45. *The Evaporating and Distilling Plant.*—There are four horizontal evaporators located in a compartment on the berth deck just forward of the engine hatch. Inside the engine hatch are two straight-tube distillers, placed a few inches higher than the tops of the evaporators. For make-up feed, running on the main condensers, the four evaporators are rated at 38,000 gallons for 24 hours. One evaporator will usually supply water for ship's use.

46. This plant should be sufficient to give the necessary make-up feed and supply distilled water for the ship. But its efficiency is lessened by the tendency of the evaporators to foam, not only when on the condensers for make up, but when running on the distillers for drinking water. In many cases this foaming is probably due to neglect on the part of the evaporator men, since the evaporator room is so hot in the tropics that no man can stand it for any length of time. There is a good supply of fresh air coming into this room from a blower, but it is not properly distributed in the room, and most of it escapes at once by the upcast ventilator, leaving the air on the sides of the room unchanged.

47. The Board recommends that the distillers be placed higher up in the engine-room hatch, so as to give a rise to the steam pipes from the evaporator, both for the distillers and for the condensers.

48. The Board recommends that new coils for the evaporators, with composition heads instead of cast-steel heads, be furnished and fitted. The present steel heads give trouble by leaking, and

any slight quantity of oil in the steam, from the Niclausse grease used on the tube fittings of the boilers or from other causes, passes into the shell of the evaporator and appears in the water. Baffle plates should also be fitted in the evaporators. There should also be furnished one or two spare coils, with composition heads, for the evaporators. The Board recommends removable sheet-metal casings for the heads of the evaporators, in place of the present unsightly canvas covers.

49. Of such great importance is this question of the evaporator plant of any ship fitted with water-tube boilers, that the Board feels called upon to offer a few general remarks on the subject.

50. The Board is of the opinion that in future the entire arrangement of the plant on board ship for making fresh water for the boilers, and also the arrangement of the pumps for feeding this water to the boilers, must be radically changed, if water-tube boilers of any design are to be a success.

51. Sufficient space will have to be given to the evaporators, in order that a type of evaporator best suited for the work may be installed. The present type of evaporator has been evolved to suit the limited head room in the space hitherto assigned, and it is probably not the best type. In the future the evaporator room on ship board must suit the evaporator, and the space allotted must be such that men may stand regular watch there, with no more discomfort than is suffered by those who stand watch in the engine room. Evaporators should not again be crowded into places so hot that men cannot give them the attention they demand. Unless evaporators are closely watched, especially when running on the condensers for make-up, they will foam and salt water will appear in the boiler feed. The D. B. compartments used for reserve-feed tanks should have but one suction pipe leading into them, and that pipe from a pump that handles fresh water only, so that no chance opening of a sea-suction valve shall be able to spoil the reserve feed by a dose of salt water. Any pump intended to pump water into a boiler should never be piped for any other purpose whatever. This should be a rigid rule, so that there may be no loss of feed water through leaky valves, nor any chance of a suction pipe full of greasy or salty water going into the boilers.

52. This problem of providing fresh water, and of keeping fresh water and salt water apart on board ship, is of the very first importance. It is of more importance than the question of what type of boiler to use, for all types of water-tube boilers depend upon the solution of this problem for their existence.

53. The Board is of the opinion that this subject of the best method of making pure fresh water for the boilers, and of maintaining its purity and freshness when once made, is of such importance that it is well worthy of investigation by a Board specially appointed for that purpose. This Board could take up the question of the best type of evaporator and distiller for use on board ship. This Board could also take up the question of main condenser, whether two main condensers should be used in each engine room, and the auxiliary condensers dispensed with. Also the best type of grease extractor, and its location in the feed line, and what further precautions should be taken to keep oil out of the feed water.

54. The general rules such a Board would lay down for the improvement of the evaporator plant on board ship would probably add weight, but it is believed that such additional weight should be given for this important object. Part of this additional weight might be made up by a reapportionment of the present machinery weights looking to the omission of unnecessary pipes, valves and fittings. The space required for the re-arranged plant should be insisted upon in all new designs.

55. *Remarks and Conclusions on the Performance of the Boilers.* The Board finds that the boilers of the *Maine* worked well during the voyage to Culebra and return. They accepted, without apparent damage, the treatment that boilers of any kind usually receive in service on board ship. At times the feed water was brackish and oily, and at all times the boilers were fired by men of little experience with steam generators of this type. The water level was easily maintained, and the water was quiet in the glass. On the trip south there was difficulty in keeping up the speed for 115 revolutions of the main engines; but on the return trip, the good results of placing an acting chief water tender in charge of all the fire rooms was shown by the fact that steam was easily maintained for 120 revolutions of the engines. The Board be-

lieves that the boilers of the *Maine* are fully equal to the work they were designed to do, and the Board sees no reason why they should not continue to do this work, provided they are given the care necessary for the efficiency of any water-tube boiler.

56. *Bending of Tubes.*—Examination of the boilers after the return of the ship to Newport News showed that practically all tubes of the two bottom rows were bent more or less, but none bent enough to require renewal. This bending is due to the contraction of the lower part of the tubes when the furnace doors are opened for coaling the furnaces, and will, of course, be more marked in boilers that have been the hardest fired. With a fixed rate of combustion it is probable that the tubes will bend a certain amount and then bend no more, until the rate of combustion is increased. This shows the necessity for a fixed and uniform rate of combustion in port and at sea.

57. *Fresh Water.*—From the experience with the boilers of the *Maine* on this voyage the Board concludes that while clean, fresh water is necessary for the life and efficiency of any water-tube boiler, it is no more a necessity for the Niclausse boiler than for any other boiler of the large-tube variety.

58. *Limit of Capacity.*—The boilers of the *Maine*, like all water-tube boilers of this general type, have a limit of capacity that cannot be exceeded without danger of reversal of the current of circulation with consequent burning of tubes. Exactly what the limit of capacity of these boilers is, the Board had no way of finding out. Mr. Sneddon, of the Stirling Boiler Works, states that experiment had shown that the limit for the boilers of the *Maine* is reached when enough coal is burned on the square foot of grate per hour to evaporate 300 pounds of water per hour under service conditions. This limit applies to these boilers only, since a different proportion of the two parts of the headers, and larger drums, will raise the limit of capacity. Assuming that one pound of coal will evaporate eight pounds of water in the boilers of the *Maine*, under service conditions, the limit of capacity would be when 37.5 pounds of coal are burned per square foot of grate per hour. The Board believes that efficient firemen, with the natural draft of the high smoke pipes, provided the ventilators are raised as recommended, can obtain a rate of com-

bustion in these boilers that will supply all the steam the main engines and auxiliaries can use, and that this rate of combustion will be below the limiting rate found above. From this it follows that forced draft is unnecessary and undesirable for these boilers as installed in the *Maine*, and the Board recommends that forced draft never be used with them, but that any increase of power for the engines should in all cases be obtained by adding to the number of boilers in use. This recommendation is not intended to prohibit the use of the blowers with open fire room for ventilating purposes. The fire-room ventilators should be carried higher so as to clear the boats stowed in the cradle. This would give improved natural ventilation for the fire rooms, and would permit the use of the blowers for ventilating without having hot air from the fire-room hatches blown back into the fire rooms.

59. *More Recent Types.*—The Board's conclusion that forced draft should not be used with the boilers of the *Maine* needs not, and does not, apply to the same type of boiler differently proportioned. In the newer boilers of this type building for the Government the proportions are such that the limit of capacity is probably beyond the reach of the rate of combustion that will be attained with forced draft. The Board recommends that one boiler of those now building for the Government at the Stirling Works be set up at the works and tested under a forced draft strong enough to give a rate of combustion higher than will ever be required of them in service. This test will show whether the limit of capacity is likely to be reached under service conditions. The test to be witnessed by a Board of officers detailed by the Department.

60. *Personnel.*—These boilers of the *Maine* are a fine example of mechanical work, and to keep them in proper order will require the services of men of intelligence and mechanical skill. The Board was impressed with the obvious fact that the quality of the personnel is of more importance than the quantity. The water-tube boiler calls for more brain work, and at the same time less physical effort on the part of the fireman. To get this quality of fireman he will have to be specially trained. The custom on board the *Maine* now is that when necessary to take out

tubes the firemen and water tenders take them out, while the machinists and boiler makers put them in. With the experience now acquired the men have become very proficient. The average time to remove a tube and replace one is about fifteen minutes, provided the generating tube does not turn with the effort required to unscrew the lantern plug. A suitable tool is provided to hold a turning tube.

61. Taking up in order the four points the Board was especially directed to report upon:

(*a*) *The Water Level.*—The water was carried at about the same height in all the boilers, from ½ to ¾ glass. The water level was quiet, and there was no appearance of spasmodic circulation of the water when the main engines were stopped and started, or when the bottom blows were used. One water tender looked out for four boilers, but this is about all that one water tender can do. The isolation of the eight fire rooms, the only communication being through narrow passages, barely wide enough for a man to squeeze through, makes it very difficult to obtain uniform firing and water tending, so essential to the efficient management of water-tube boilers.

(*b*) 62. *Automatic Devices.*—The automatic feed devices were in use, but the Board is of the opinion that they cannot be trusted. The water tenders would often report them as working well, but it was evident that no water tender trusted these devices to regulate the height of water for him, but that all of them preferred to give the slight occasional turn of the check-valve wheel required to accomplish this result. On the last day of the run it was reported that only seven of the twenty-four were working. The Board believes these devices unnecessary, and recommends that they be taken off. The removal of this casting would make the feed checks more accessible than they now are. As to the automatic devices for shutting off the water glasses in case of breakage, there is but one feeling about them, and that is a feeling of relief that they are absent. The gauge glasses in use on this ship do not last well. The glasses are too long, and the fittings too light. It is a common occurrence for several glasses to break during a watch. The Board recommends that a more trustworthy type of water gauge be fitted.

(*c*) 63. *Firing.*—As stated before, the fires were not carried as they should have been, especially on the run south. The firemen seemed unable to forget that they were not firing Scotch boilers. There was no one in the fire rooms to give the times for firing and to regulate the length of time the operation of firing and of leveling the fires should take. On the trip north, at the suggestion of the Board, a water tender, with supervising authority over all fire rooms, was detailed for each watch. The result was a very marked improvement in the firing and water tending.

(*d*) 64. *Visibility of Water Glasses.*—The water gauges of each fire room are now visible from all parts of the fire room where the water tender usually is in the discharge of his duty. To make these more fully visible yet, the Board recommends that the lagging be taken off all over-head feed pipes in the fire rooms. The valves in the fire rooms are conveniently placed, and the steam gauges are plainly visible. The fire rooms are roomy and convenient, and they are fairly well ventilated. By carrying to a greater height certain ventilators now partly blanked by the boats stowed in the cradles, the fire-room ventilation will be improved. In connection with the subject of convenience of fire rooms, the Board calls special attention to the inadequate space allotted to the boiler-feed pumps, both main and auxiliary. These important pumps are thrust into little recesses cut out of the bunkers, with not sufficient space around them for proper maintenance and repair.

65. For the proper care of these boilers a supply of air under pressure is necessary. At present the necessary air for emptying tubes, etc., is obtained from the torpedo compressor, but this compressor is not equal to the double duty required of it. The Board recommends that six Westinghouse air compressors, of 40 cubic feet capacity each per minute, be installed.

Respectfully submitted,

J. A. B. SMITH, *Captain, U. S. N.*
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W. M. PARKS, *Lieutenant Commander, U. S. N.*

The SECRETARY OF THE NAVY,
Washington, D. C.

EXPERIMENTAL TANKS.

Attention has been called by Sir Nathaniel Barnaby to the "understanding" that the Institution of Naval Architects would take measures for getting from shipowners and shipbuilders the money required for building an experimental tank as an addition to the National Physical Laboratory. The sum required is £15,000. It has not been forthcoming. Sir Nathaniel is exercised in his mind about this recalcitrance, and writing to the "Times" on the subject, he explains the delay in the following words: "Shipowners will not subscribe until they see that shipbuilders are ready to do so. Shipbuilders hesitate because they either doubt the value of the method of research, or they consider that, if it is valuable, their own establishment should have an installation of its own. Perhaps they do not quite realize how much of the success of such an establishment depends on the special education and training of the observer."

At the time when the proposal to establish such a tank was first put forward, it was objected that Bushey House was not the proper place for it, and it was argued, as we think in a wholly unanswerable fashion, that there was no special reason of any kind why an experimental tank should be constructed far inland, while, assuming that it is really wanted, the proper place was at some large shipbuilding center. The Clyde already has one at Dumbarton—the result of the private enterprize of Messrs. Denny Brothers. The Admiralty have two in the south at Chelston Cross and Haslar. Liverpool, or Hull, or Newcastle-on-Tyne are all three desirable localities for one or more tanks. Nothing at all can be advanced from a shipowner's or a shipbuilder's point of view in favor of London. Sir Nathaniel Barnaby is evidently in possession of this fact, for he adduces the only reason available for incorporating the tank with the National Physical Laboratory,

when he says by implication that a tank erected elsewhere will not possess the advantage of being worked by men who understand it.

We venture to say that Sir Nathaniel Barnaby will have much difficulty in making out a case for a tank at Bushey on this basis. We shall even go so far as to say that if Sir Nathaniel will make inquiries he will find that it is just because shipbuilders have small faith in the national science men, and a great deal in their own ability to do their own work, that the funds have not been subscribed long since. We are ourselves entirely at a loss for a reason why Bushey men should possess any special fitness for such work as compared with numbers of men who could be selected from our shipyards as peculiarly qualified for working an experimental tank to the best advantage. The scheme as it stands is, rightly or wrongly, mixed up with South Kensington, and the past history of South Kensington has not made it acceptable to the leading manufacturers of the day. It will be found very difficult, if not impossible, to persuade either shipowner or shipbuilder that £15,000 spent on a tank at Bushey will be spent to the best advantage. Possibly the difficulty might be got over if the names of the tank staff, being published, proved convincing. As the affair stands no one will subscribe to the construction of the tank until he knows by whom it will be used; and no staff can be nominated until the tank is at all events being constructed. A deadlock is so far the result of the dilemma.

There is, moreover, another difficulty which Sir Nathaniel fully appreciates. He has appended to his letter published in the "Times" of Saturday a long communication from the Admiralty in praise of tanks, and explaining the nature of the valuable results obtained from their use. Sir Nathaniel sees that the experimental tank is regarded with some doubt as regards its practical utility. We think that it must be admitted that this doubt is justified from the point of view of the ordinary shipowner and the men who build ships for him. An enormous number of ships engaged in trade never exceed a speed of 12 knots. The regular rate of the ocean tramp is nearer 10 knots.

The power required to propel her at low speeds is very minute—it has been calculated that the heat produced by burning an envelope would suffice to carry a ton of cargo half a mile. We need not go so far as this, and yet see that no tank experiments can help the shipowner much. Indeed, the shape of his boats is, in a way, determined for him by conditions which have nothing to do with speed or hull resistance. The value of the tank is experienced when very high velocities are to be attained, and the utmost possible speed has to be got with any given power.

For work of this kind the tank, properly used, can teach valuable lessons. But the number of high-powered ships is comparatively small, and the firms building them are of a dignity and importance which will prevent them from going hundreds of miles to Bushey for information. If they think that tanks are worth the cost of them they will have them, just as Messrs. Denny have one at Dumbarton. The utmost, it appears to us, that a staff with a tank at Bushey could do is carry out research in the abstract. How far the results obtained would be of use in practical shipbuilding and marine engineering must for the time being remain an open question.

It is idle to deny that a widespread doubt lingers in our great shipping centers as to the trustworthiness of experiments made with models. After all, sea-going conditions cannot be reproduced, even in miniature, with certainty in a tank. *Shamrock III* was the result, we believe, of tank trials. Whether the American boat was or was not, we are not certain; we believe that Herreshoff designed her without help from a tank. But that is a small matter. The Dumbarton tank undoubtedly gave most perfect lines for the Channel turbine steamer *The Queen*, and not a few other boats; but beyond this few competent authorities are disposed to go.

The Admiralty claim for the government tanks that they have supplied valuable information about propellers. It appears to us that this is just the department of investigation in which the tank breaks down. We must either assume that the teachings of the government tanks have been wholly ignored, or else that they have been entirely misleading in such cases as that of the

Hyacinth and *Minerva*, to say nothing of other ships in the navy. Notwithstanding the existence of the Torquay tank for many years, no one now disputes that at this moment the designing of screw propellers is almost entirely glorified guesswork; and no one really can tell at the outset whether any given type or dimensions of a propeller are the best that it is possible to adopt for any particular ship. It is probable that all tank experiments on screw propellers are untrustworthy, and must always remain so, because it is impossible to reproduce in a tank the conditions prevailing in the open sea.

We are quite prepared to advocate the possession of an experimental tank at every port where high-powered steamers are constructed. We believe that they can be made worth the outlay upon them, in so far as they supply a particular kind of information that cannot be obtained otherwise. But we also believe that their value may be over-rated by enthusiasts, and we are certain that a tank in connection with the National Physical Laboratory will not content shipbuilders or shipowners. The fact is forgotten that those interested in experiments, from commercial reasons, are always desirous to keep the results obtained for their own use and benefit. It would be extremely difficult to do this with a public tank. It is for this reason, among others, we fancy, that subscribers will not come in; and, after all, the Bushey tank, if it is ever made, will, as we have said, have to carry out abstract research. It remains to be seen whether the shipowners or shipbuilders can be induced to provide facilities for such work. The cost is, after all, so small that the funds ought to be forthcoming. But we can assure Sir Nathaniel Barnaby that the best way to obtain them is to establish confidence in the Royal Society, and its fitness to carry on the work.—"The Engineer,"

London, Sept. 18, 1903.

Errata.

Page 1004, 2d line, read d_A

Page 1016, 3d line, read L^2_c

Page 1023, 15th line, read L^2_c

Page 1035, 12th line, read $\frac{\Sigma SL}{\frac{6}{2}}$

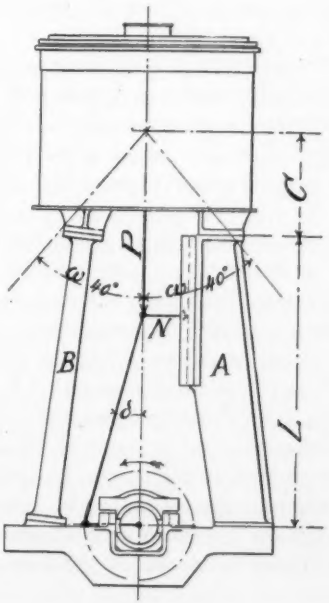
Page 1036, in equation 17, read $\frac{\frac{\Sigma S}{6}L^3}{3}$

INVESTIGATION OF THE STRESSES ON FRAMES OF VERTICAL STEAM ENGINES—AS MODIFIED FOR MARINE ENGINES.

By John H. Meyer, M. E., Associate Member.

The frames of marine steam engines as installed on board ship have to withstand severe stresses due to the rolling and pitching of the vessel. These stresses must be considered in addition to the stresses due to the steam pressure upon the piston and its component upon the guides as shown in Fig. 1.

Fig. 1



The force P on the piston of a vertical inverted steam engine acts always in a direction perpendicular to the base of the engine, and produces alternately a tensile and a compressive stress upon the housing A and column B, while a bending stress, acting in the plane of the crank path, is produced by the normal force N acting upon the guide.

The rolling motion of the ship being in the athwartship direction, therefore the stresses on the frames due to the rolling of the ship must act in the plane of the crank path, or very nearly in this plane. The maximum rolling angle is approximately 40 degrees to either side, and the force acting upon the frames of the engine, produced by the rolling of the ship, is the moment of the weight W of the cylinder, including the piston, piston rod and crosshead, acting at the center of gravity, with its components upon the frames A and B. The component $W \sin 40°$ produces a bending moment acting at a distance $L + C$ (see Fig. 1), while the component $W \cos 40°$ produces a compressive stress.

The pitching motion exceeds only in very extreme cases 10 degrees from the horizontal line in a fore-and-aft plane; the force producing a bending stress upon the frames is again ΣW acting with the component $\Sigma W \sin 10°$ in a plane parallel, or very nearly so, to the vertical plane through the center line of the shaft.

A certain class of naval vessels, in which the ability to ram is considered a part of their fighting qualifications, will produce an additional bending stress upon their engine frames while in ramming action. The force which produces the bending stress upon the frames is the inertia of the weight of the cylinders acting in the same direction as the pitching stress, that is in the fore-and-aft plane of the ship. An air pump direct-connected to the engine also produces another bending stress on the frames if the pump beam bearings are carried by the frames, but generally, on large modern marine engines, the air pump is independent, or the pump-beam bearings are carried in such a manner that the bending stress on the frames produced by them is nothing, or very nearly so.

The above mentioned bending stresses on the frames in different directions produce tensile and shearing stresses on the bolts securing the frames to the bed plate and to the cylinder respectively. It requires extra-strong securing bolts to withstand these stresses in combination with the stresses due to the steam pressure upon the piston.

The frames of modern naval engines are generally designed for light weight, and are therefore built of forged-steel columns in pairs on each side, and fitted with diagonal rolling stays or tie rods. These tie rods should be so designed as to relieve the columns from all side stresses, so that the columns themselves will only have to carry the weight of the cylinders and withstand the stress caused by the steam pressure upon the piston. Sometimes, however, it is necessary to arrange the tie rods in a less efficient manner, but then the securing bolts generally become subject to the before-mentioned shearing stress, and the columns are required to withstand an additional bending stress. The frames of vertical inverted marine engines for the merchant marine are designed, according to the taste of the designer, as housing on each side, single or in pairs, for each cylinder, or as one housing on the side of the crosshead guide, if a single guide is preferred, and one or two columns on the opposite side of the cylinder axis for one cylinder. This latter system is also adopted by some builders of naval engines, but in the U. S. naval service the design with the frames of the forged-steel column type is preferred.

The greatest stress upon the frames occurs when the rolling is in the direction towards the guide, or, in other words, when the rolling force acts in the same direction as the force N (see Fig. 1.) Throughout the investigation this direction of motion is called the ahead motion. It follows, the housing A is always under the stress of the force N with the engine in ahead motion, and that while the greatest stress upon the frames occurs with the engines going ahead and the vessel rolling towards N, it is clear that the stress must be reduced materially when the ship is going ahead and rolling away from N.

Engines provided with double guides, that is with the backing

guide on the front frame, will have to withstand the same stress on the frames if the motion of the engine is reversed and the rolling is towards the front of the engine, because the direction of the rolling force is identical again with the direction of the force N exerted on the guides in backing.

If an air pump is attached to the engine in such a manner as to produce a stress on the frames, this stress will be always in the same direction, no matter in which direction the engine is turning. Now, since the center of gravity of the weights of the cylinder, etc., is at the distance $C + L$ (Fig. 1), from the lower face of the frames A and B, we have at the top end of A and B an equivalent weight necessary to produce the same effect $= \frac{W.(L+C)}{L} = W_1$ and $W_1 \sin \omega =$ the force acting at the top end of the housing A as well as of column B.

Before proceeding further it is necessary to state that this investigation is carried out on similar lines to those followed by Mr. G. Schwarz, Obertuerkheim Stuttgart, in "The investigation of stresses on frames of vertical inverted stationary steam engines," translated by me and reprinted in the February number of this year's JOURNAL of the Society.

This method of Mr. Schwarz is, in my opinion, the best published on the subject. Therefore, I have modified the formula and equasions of this method to make it applicable to the investigation of stresses on the frames of marine engines of similar type.

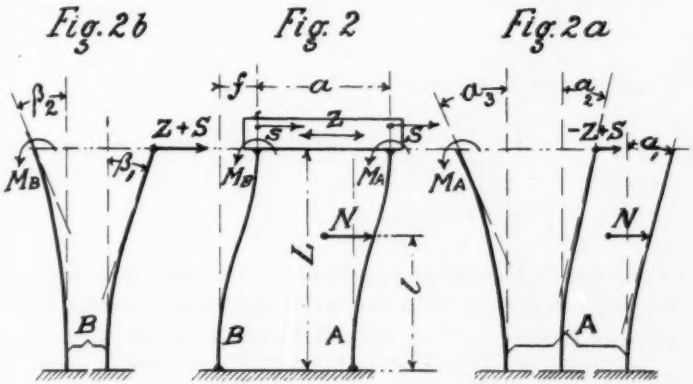
Now, it is assumed again, that the lower ends of the frames are fixed to the bed plate; the ends connected to the cylinder are treated as being secured to a crossbar of sufficient strength to prevent its undergoing any change in shape by the action of any stress; further, that the frames have a vertical position and that their moments of interia J_A and J_B are alike for every section over their entire length. It must be borne in mind that, if the column B or housing A are in pairs for one cylinder, the sum of the moments of inertia are always J_A and J_B respectively.

With the assumed connection of the top ends of the frames,

it can be taken for granted that the stresses on the frames due to the rolling motion will be equal at both housing A and column B, therefore the rolling force on each of these frames will equal

$$\frac{W_1}{2} \sin \omega = S.$$

The different forces are now applied as shown in diagram (Fig. 2).



The force N acts at a distance l from the lower end and tends to turn the housing A to the right. The force S acts in the same direction, but the force Z in the opposite direction, both at the distance L, therefore the difference of the latter two forces at the distance L tends to turn the housing A to the right. The moment M_A, also at the distance L, turns it to the left. Upon the column B is acting the sum of the forces $Z + S$ tending to turn the column to the right, while the moment M_B has the tendency to turn it to the left. All three of these forces are acting at the distance L.

The angle β_1 of the tangent to the elastic line at the upper end of column B, see (Fig. 2b), due to the sum of the forces $Z + S$ is, with E_B as a constant modulus of elasticity, =

$$\beta_1 = \mp \frac{ZL^2}{2J_BE_B} + \frac{SL^2}{2J_BE_B} *$$

and the angle β_2 of the tangent, due to the bending moment M_B, at the same end,

$$\beta_2 = \frac{M_BL}{J_BE_B}$$

and as the assumed crossbar connection at the top end cannot change its shape, if the top ends of the frames are fixed to a properly designed cylinder, we must have

$$\beta_1 - \beta_2 = 0$$

or substituing the values of β_1 and β_2

$$\left(\mp \frac{ZL^2}{2} + \frac{SL^2}{2} - M_BL\right)\frac{1}{J_BE_B} = 0;$$

therefore

$$M_B = \mp \frac{ZL}{2} + \frac{SL}{2} \quad \ldots\ldots\ldots \quad (1)$$

Using the same reasoning, we obtain for the upper end of housing A the angle of the tangent to the elastic line due to the action of forces $(-Z+S)$, N and M_A (see Fig. 2a.)

$$a_1 = \mp \frac{Nl^2}{2J_AE_A}$$

$$a_2 = \pm \frac{ZL^2}{2J_AE_A} + \frac{SL^2}{2J_AE_A}$$

$$a_3 = \frac{M_AL}{J_AE_A}$$

and further, since

$$a_1 + a_2 - a_3 = 0$$

we find, by substitution

$$M_A = \mp \frac{Nl^2}{2L} + \frac{SL}{2} \pm \frac{ZL}{2} \quad \ldots\ldots\ldots \quad (2)$$

*(The lower sign of + or — to be used in all following equations until further explanation.)

From equation

$$\left(\mp\frac{ZL^2}{2}+\frac{SL^2}{2}-M_BL\right)\frac{1}{J_BE_B}=0$$

by inserting the value of M_B is obtained the equation

$$\frac{1}{J_BE_B}\left[\mp\frac{ZL^2}{2}+\frac{SL^2}{2}-\left(\mp\frac{ZL}{2}+\frac{SL}{2}\right)L\right]=0=\frac{dy}{dx}=\frac{df}{dL}$$

and, after integration, we have the deflection of column $B=$

$$\frac{1}{J_BE_B}\left[\mp\frac{ZL^3}{3}+\frac{SL^3}{3}-\left(\mp\frac{ZL}{2}+\frac{SL}{2}\right)\frac{L^2}{2}\right]=f,$$

which after reduction takes the form

$$\frac{\mp ZL^3+SL^3}{12J_BE_B}=f \text{ of column } B. \quad \cdots \quad (3)$$

Again, from the fixed length of connection at the top ends of the frames, the deflection f of the upper end of the housing A must be equal to f that of housing B.

The deflection of the housing A is found from equation

$$a_1+a_2-a_3=0$$

and by inserting the value of M_A. This insertion gives the following equation.

$$\frac{1}{J_AE_A}\left[\mp\frac{Nl^2}{2}+\frac{SL^2}{2}\pm\frac{ZL^2}{2}-\left(\mp\frac{Nl^2}{2L}+\frac{SL}{2}\pm\frac{ZL}{2}\right)L\right]=\frac{dy}{dx}=\frac{df}{dL}.$$

By integration and transformation, the deflection f of the top end of the housing $A=$

$$\frac{1}{J_AE_A}\left[\mp N\left(\frac{l^3}{3}+\frac{Ll^2}{2}-\frac{l^3}{2}-\frac{Ll^2}{4}\right)+\frac{SL^3}{3}-\frac{SL^3}{4}\pm\frac{ZL^3}{3}\mp\frac{ZL^3}{4}\right]$$
$$=f=\mp\frac{ZL^3+SL^3}{12J_BE_B}.$$

Forming a new equation from these two, both being equal, we obtain

$$\frac{1}{J_A E_A}\left[\mp N\left(\frac{l^2 L}{4}-\frac{l^3}{6}\right)+\frac{SL^3}{12}\right]-\left(\frac{SL^3}{12}\right)\frac{1}{J_B E_B}=\mp\left(\frac{ZL^3}{12}\right)\frac{1}{J_B E_B}$$
$$\mp\left(\frac{ZL^3}{12}\right)\frac{1}{J_A E_A}. \quad . \quad . \quad . \quad (3a)$$

and further

$$Z=\frac{[N(3l^2L-2l^3]J_B E_B+SL^3(\mp J_B E_B \pm J_A E_A)}{(J_A E_A+J_B E_B)L^3}. \quad . \quad (4)$$

From equation (4) it is seen that the force S has no influence upon the force Z, as long as $J_A E_A = J_B E_B$, or, in other words, as long as the column B and the housing A are of the same size and material, because the value SL^3 disappears in this equation. As soon as there is a difference between $J_A E_A$ and $J_B E_B$ the value of Z will be different.

The equations (1 to 4) are obtained with the assumption that the crosshead guides are supported over their entire length by the housing A as shown in Fig. 1, but if the crosshead guide is supported at both ends only, as shown in Fig. 3, the force N will be resolved into two different forces, N^1, acting at the top end of housing A, and N_1, acting at a distance l from from the lower end of housing A. The stress on, and the deflection of housing A must change, consequently, also the equations for obtaining the relative results. These equations will be noted with sub number (1).

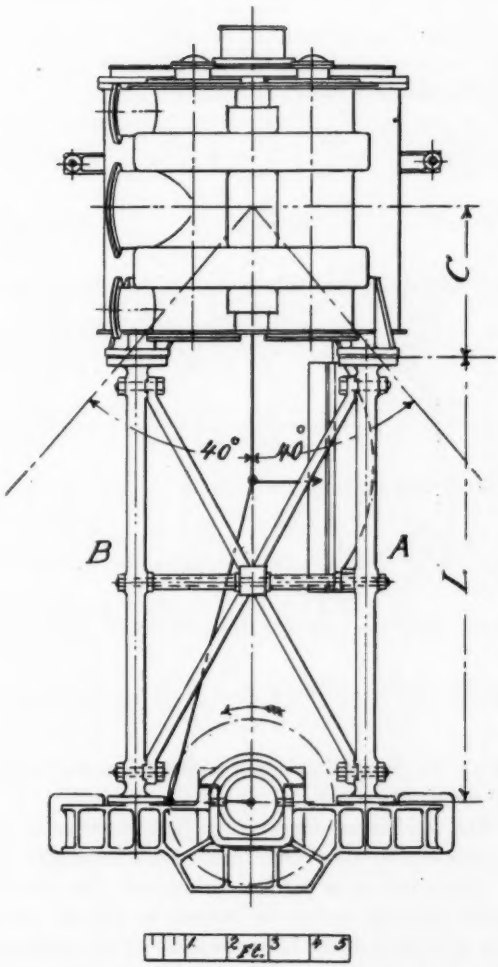
The equation for β_1 and β_2 will not change in reading, but the values of Z for β_1 and M_B for β_2 will be different (see Fig. 2). The same change, only, will appear in equation (1), but for housing A, we will have

$$a_1 = \mp\frac{N_1 l^2}{2J_A E_A}$$

$$a_2 = \pm\frac{ZL^2}{2J_A E_A}+\frac{SL^2}{2J_A E_A}\mp\frac{N^1 L^2}{2J_A E_A}$$

$$a_3 = \frac{M_A L}{J_A E_A}$$

Fig. 3



$$a_1 + a_3 - a_2 = 0$$

$$M_A = \mp \frac{N_1 l^2}{2L} \mp \frac{N^1 L}{2} + \frac{SL}{2} \pm \frac{ZL}{2} \quad . \quad . \quad . \quad (2_1)$$

$$f = \mp \frac{ZL^3 + SL^3}{12 J_B E_B} \quad . \quad . \quad . \quad . \quad . \quad . \quad (3_1)$$

deflection of column B, and

$$\frac{1}{J_A E_A}\left[\mp \frac{N_1 l^2}{2} \mp \frac{N^1 L^2}{2} + \frac{SL^2}{2} \pm \frac{ZL^2}{2} - \left(\mp \frac{N_1 l^2}{2L} \mp \frac{N_1 L}{2} + \frac{SL}{2} \pm \frac{ZL}{2}\right) L\right]$$

$$= \frac{dy}{dx} = \frac{df}{dL}$$

and after integration the deflection f of housing $A =$

$$f = \frac{1}{J_A E_A}\left[\mp \frac{N_1 l^3}{3} \mp \frac{N_1 l^2}{2}\left(L - l\right) \mp \frac{N^1 L^3}{3}\right.$$

$$\left. + \frac{SL^3}{3} \pm \frac{ZL^3}{3} \pm \frac{N_1}{2}\left(\frac{Ll^2}{2}\right) \pm \frac{N^1 L^3}{4} - \frac{SL^3}{4} \mp \frac{ZL^3}{4}\right] = \frac{\mp ZL^3 + SL^3}{12 J_B E_B}.$$

From these two equations we obtain

$$\frac{1}{J_A E_A}\left[\mp N_1\left(\frac{l^2 L}{4} - \frac{l^3}{6}\right) \mp \frac{N^1 L^3}{12} + \frac{S^3 L}{12}\right] - \frac{SL^3}{12 J_B E_B} =$$

$$\mp \left(\frac{ZL^3}{12}\right) \frac{1}{J_B E_B} \pm \left(\frac{ZL^3}{12}\right) \frac{1}{J_A E_A} \quad . \quad . \quad (3a_1)$$

and

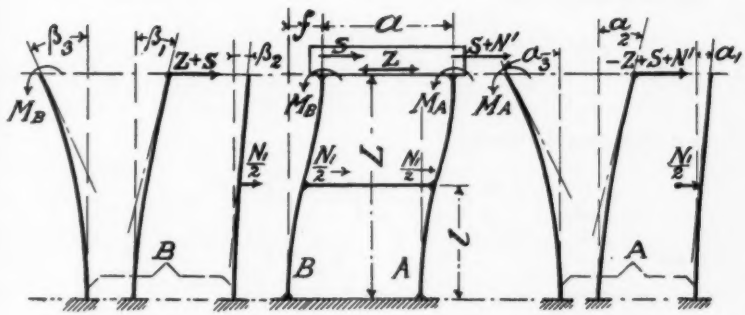
$$Z = \frac{[N_1(3l^3 L - 2l^3) + N^1 L^3] J_B E_B + SL^3(\mp J_B E_B \pm J_A E_A)}{J_A E_A + J_B E_B) L^3} \quad . \quad (4_1)$$

The value SL^3 disappears again under the same conditions as explained for equation (4).

Very often the frames A and B are tied together, as shown in (Fig. 4), particularly when both frames are housings and both provided with guide faces for the crosshead; but this tieing is also used for forged frames, as shown in Fig. 3. This connecting tie is located at the lower support of the guide face. By this means the column B and housing A are each required to

take an equal share of the force N_1 upon the lower end of the guide. The bending moment produced by this force upon column B, as well as upon housing A, is $\frac{N_1}{2}l$, consequently the bending stresses, deflection, etc., are changed again. The respective equations and formulas will be noted by sub number (2). The values, the direction of and the point of action of the forces upon housing A are shown in Fig. 4a, and upon column B in Fig. 4b.

Fig. 4b Fig. 4 Fig. 4a



Now we have, for column B, the angles of the tangents

$$\beta_1 = \mp\frac{ZL^2}{2J_BE_B} + \frac{SL^2}{2J_BE_B} \qquad \beta_2 = \mp\frac{\frac{N_1l^2}{2}}{2J_BE_B} \qquad \beta_3 = \frac{M_BL}{J_BE_B}$$

and $$\beta_1 + \beta_2 - \beta_3 = 0$$

further

$$M_B = \mp\frac{ZL}{2} + \frac{SL}{2} \mp \frac{\frac{N_1l^2}{2}}{2L} \quad . \quad . \quad . \quad . \quad . \quad (1_2)$$

For the housing A there is the same reading of the equations for the angles of the tangents as in the equations marked sub number (1) with the single exception that in the equation for a_1,

the value $\frac{N_1}{2}$ takes the place of N_1. For M_A we have

$$M_A = \mp \frac{\frac{N_1}{2} l^2}{2L} \mp \frac{N^1 L}{2} + \frac{SL}{2} \pm \frac{ZL}{2} \quad . \quad . \quad . \quad (2_2)$$

From $$\beta_1 + \beta_2 - \beta_3 = 0$$

we derive by inserting the value of M_B

$$\frac{1}{J_B E_B}\left[\mp \frac{ZL^2}{2} + \frac{SL^2}{2} \mp \frac{\frac{N_1}{2} l^2}{2} - \left(\mp \frac{ZL}{2} + \frac{SL}{2} \mp \frac{\frac{N_1}{2} l^2}{2L} \right) L \right] = \frac{dy}{dx} = \frac{df}{dL}$$

and after integration

$$\frac{1}{J_B E_B}\left[\mp \frac{ZL^3}{3} \pm \frac{ZL^3}{4} + \frac{SL^3}{3} - \frac{SL^3}{4} \mp \frac{\frac{N_1}{2} l^3}{3} \mp \frac{\frac{N_1}{2} l^2}{2} (L - l) \pm \left(\frac{\frac{N_1}{2} l^2}{2} \right) \frac{L}{2} \right] = f$$

and after transformation

$$\frac{1}{J_B E_B}\left[\mp \frac{N_1}{2} \left(\frac{l^2 L}{4} - \frac{l^3}{6} \right) + \frac{SL^3}{12} \mp \frac{ZL^3}{12} = f. \right. \quad . \quad . \quad (3_2)$$

From $$a_1 + a_2 - a_3 = 0$$

by substituting their values and inserting the value of M_A we have

$$\frac{1}{J_A E_A}\left[\mp \frac{\frac{N_1}{2} l^2}{2} \mp \frac{N^1 L^2}{2} + \frac{SL^2}{2} \pm \frac{ZL^2}{2} - \left(\mp \frac{\frac{N_1}{2} l^2}{2L} \mp \frac{N^1 L}{2} + \frac{SL}{2} \pm \frac{ZL}{2} \right) L \right] = \frac{dy}{dx} = \frac{df}{dL}$$

and, after integration, the deflection f, that of housing A is now

$$f = \frac{1}{J_A E_A}\left[\mp \frac{\frac{N_1}{2} l^3}{3} \mp \frac{\frac{N_1}{2} l^2}{2} (L - l) \mp \frac{N^1 L^3}{3} + \frac{SL^3}{3} \pm \frac{ZL^3}{3} \pm \frac{\frac{N_1}{2} L l^2}{2} \pm \frac{N^1 L^3}{4} - \frac{SL^3}{4} \mp \frac{ZL^3}{4} \right]$$

and after reduction

$$f = \frac{1}{J_A E_A}\left[\mp\frac{N_1}{2}\left(\frac{l^2 L}{4} - \frac{l^3}{6}\right) \mp \frac{N^1 L^3}{12} + \frac{SL^3}{12} \pm \frac{ZL^3}{12}\right]$$

The deflection of housing A is equal to that of column B, therefore

$$\frac{1}{J_A E_A}\left[\mp\frac{N_1}{2}\left(\frac{l^2 L}{4} - \frac{l^3}{6}\right) \mp \frac{N^1 L^3}{12} + \frac{SL^3}{12} \pm \frac{ZL^3}{12}\right] =$$
$$\frac{1}{J_B E_B}\left[\mp\frac{N_1}{2}\left(\frac{l^2 L}{4} - \frac{l^3}{6}\right) + \frac{SL^3}{12} \mp \frac{ZL^3}{12}\right]$$

Transforming this new equation, we obtain

$$\mp Z(J_A E_A + J_B E_B) L^3 = \mp (N^1 L^3) J_B E_B \mp \left[\frac{N_1}{2}(3l^2 L - 2l^3)\right] J_B E_B$$
$$\pm \left[\frac{N_1}{2}(3l^2 L - 2l^3)\right] J_A E_A + SL^3(+J_B E_B - J_A E_A) \quad . \quad (3a_2)$$

and further

$$Z = \frac{(N^1 L^3) J_B E_B + \left[\frac{N_1}{2}(3l^2 L - 2l^3)(\mp J_B E_B \pm J_A E_A) + SL^3(\mp J_B E_B \pm J_A E_A)\right]}{(J_A E_A + J_B E_B) L^3} \quad (4_2)$$

For example, thus :—

The low-pressure cylinder (Fig. 3) of a triple-expansion engine has an actual piston area of 4778.4 square inches. The initial pressure is 28 pounds per square inch.

The force N = 19,730 pounds.

" " N^1 = 8,632 "

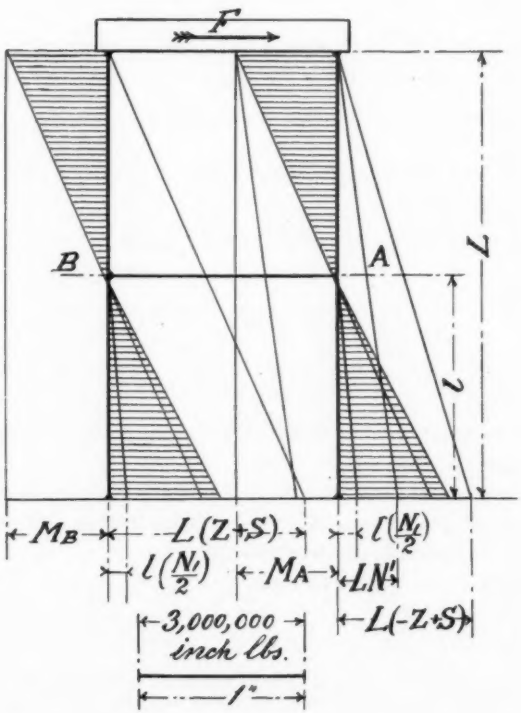
" " $\frac{N_1}{2}$ = 5,549 "

The cylinder is connected to the bedplate by four forged steel columns, two of them forming the column B, and two, with the guide face, forming the housing A.

Column B and housing A are connected as shown in (Fig. 4) and are provided with rolling ties (diagonal crosses between every athwartship pair of columns) as they have not sufficient

strength nor are they intended to withstand all stresses without side staying, but in the first calculations the rolling ties are neglected in order to ascertain how the combined bending forces act on the bar columns and what effect they will have.

Fig. 5



Each column is $5\frac{1}{4}$ inches diameter and the enlargements for the tie-rod connections are of a net section fully equal to the area of the body of the column; we therefore assume them to be of equal section over the entire length.

$J_A = J_B = 74.6$ inch^4.
$E_A = E_B = 30{,}000{,}000$.
$W = 51250$ pounds $L = 129$ inch $C = 51$ inch.

$$W_1 = \frac{129 + 51}{129} \times 51250 = 71512 \text{ pounds.}$$

$S = 22984$ pounds.
$l = 64$ inch.

Now, as $J_A E_A$ is equal to $J_B E_B$, the values $\frac{N_1}{2}$, etc. and S, etc. disappear in equation (4_2) and the amount of Z, therefore, from equation (4_2),

$$Z = \frac{8632 \times (129)^3}{2 \times (129)^3}$$

$$Z = 4316 \text{ pounds.}$$

In Fig. 5 are shown the bending moments as they appear over the entire length of column B and housing A, the amount of the different moments and the actual amount of the force F, which acts as the bending stress upon the framing, is given below:

From equation (1_2) we fine M_B.

$$M_B = \frac{4316 \times 129}{2} + \frac{22984 \times 129}{2} + \frac{5549 \times (64)^2}{2 \times 129}$$

$$M_B = 1848946 \text{ inch pounds.}$$

M_A is now:

$$M_A = \frac{\frac{N_1 l^2}{2}}{2L} + \frac{N^1 L}{2} + \frac{SL}{2} - \frac{ZL}{2}$$

$$M_A = \frac{5549 \times (64)^2}{2 \times 129} + \frac{8632 \times 129}{2} + \frac{22984 \times 129}{2} - \frac{4316 \times 129}{2}$$

$$M_A = 1848946 \text{ inch pounds.}$$

F_A contains the values:

$$F_A = \left(-Z + S + \frac{\frac{N_1 l}{2}}{L} + N^1\right) = -4316 + 22984 + 2753 +$$

$$8632 = 30053 \text{ pounds.}$$

Moment $F_A = 30053 \times 129 = 3876837$ inch pounds.

$M_A = 1848946$ inch pounds.

F_B contains the values :

$$F_B = \left(Z + S + \frac{\frac{N_1 l}{2}}{L}\right) = 4316 + 22984 + 2753 = 30053 \text{ pounds.}$$

Moment $F_B = 30053 \times 129 = 3876837$ inch pounds.

$M_B =$. 1848946 inch pounds.

Moment $(N^1)\ L = 8632 \times 129 = 1113528$ inch pounds.

$(-Z + S)\ L = 18668 \times 129 = 2408172$ inch pounds.

$(Z + S)\ L = 27300 \times 129 = 3521700$ inch pounds.

$$\left(\frac{N_1}{2}\right) l = 5549 \times 64 = 355136 \text{ inch pounds.}$$

We have on column B

$$\left(Z + S\,\frac{\frac{N_1 l}{2}}{L}\right) = F_B,$$

and on housing A

$$\left(-Z + S + \frac{\frac{N_1 l}{2}}{L} + N^1\right) = F_A$$

$$F_A + F_B = \left(2S + \frac{N_1 l}{L} + N^1\right) = 60106 \text{ pounds.}$$

The deflection which would occur is given by equation (3_2) and is

$$f = \frac{5549\left(\frac{(64)^2 \times 129}{4} - \frac{(64)^3}{6}\right) + \frac{22984 \times (129)^3}{12} + \frac{4316 \times (129)^3}{12}}{74.6 \times 30{,}000{,}000}$$

$$f = 2.401 \text{ inch.}$$

To reduce the deflection, let us say to about .024 inch, it would require a value for $J_A E_A$ or $J_B E_B$ one hundred times as large, and if we assume the frames to be steel castings for which E is 25,000,000, the multiplier for a mean J_B or J_A must be 120.

The actual $J_B = 74.6$ inch$^4 = J_A$

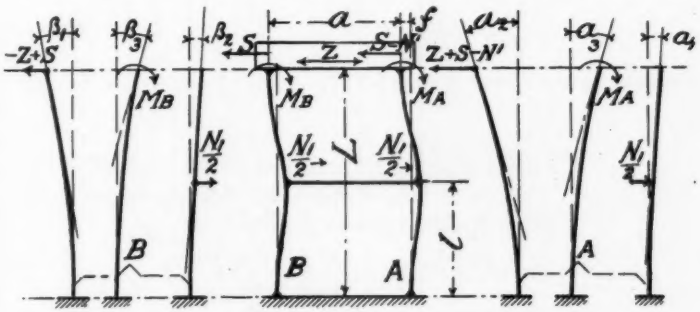
$74.6 \times 120 = 8952$ inch^4

for a housing A, and 4476 inch^4 for each of the two columns combined to form column B. For housing A would be required a mean section of a rectangular form of about 24 inches in the direction of the forces, by 18 inches wide by $1\frac{3}{8}$ inches thick, and for each of the columns representing column B a section of about 20 inches $\times$ 13 inches $\times$ $1\frac{1}{4}$ inches.

This development shows how to obtain the size of the mean section of frames from the deflection.

Assuming now that the ship is rolling 40 degrees away from the guides, while the engine is in ahead motion, it follows that

Fig. 6b Fig. 6 Fig. 6a



the rolling forces act away from the guides also as indicated in (Fig. 6). The action of the combination of the forces upon housing A is shown in (Fig. 6a), and the corresponding one upon housing B is shown in (Fig. 6b).

It will not be necessary to repeat the developments of the equations, but the results of these developments show that it is necessary to use the equations with the upper sign of + or —.

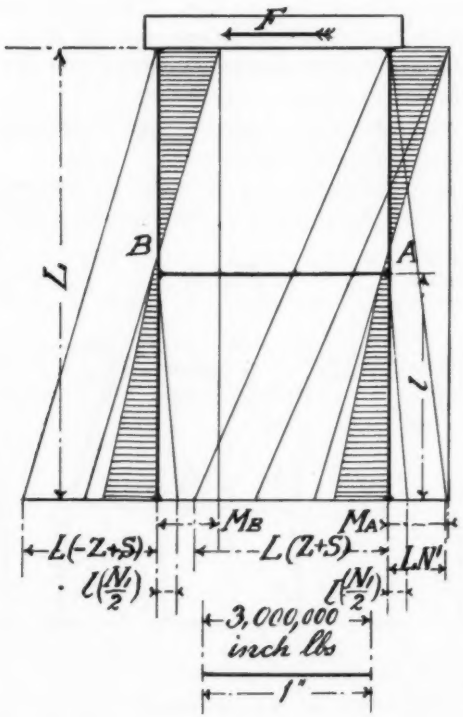
The value of Z remains the same = 4316 pounds.

The deflection would be, in this case,

$$f = \frac{-5549\left(\frac{(64)^2 \times 129}{4} - \frac{(64)^3}{6}\right) + \frac{22984 \times (129)^3}{12} - \frac{4316 \times (129)^3}{12}}{74.6 \times 30{,}000{,}000}$$

$$f = 1.1273 \text{ inch}$$

Fig. 7



The bending moments, as they appear in this case over the whole length of column B and housing A are shown in Fig. 7, and the amount of the moments and of the force F are as follows:

From equation (1_2) we find now M_B

$$M_B = -\frac{ZL}{2} + \frac{SL}{2} - \frac{\frac{N_1 l^2}{2}}{2L}$$

$$M_B = -\frac{4316 \times 129}{2} + \frac{22984 \times 129}{2} - \frac{5549(64)^2}{2 \times 129}$$

$M_B = 1115990$ inch pounds

and from equation (2_2) we get for M_A, by using the upper signs:

$$M_A = -\frac{\frac{N_1 l^2}{2}}{2L} - \frac{N^1 L}{2} + \frac{SL}{2} + \frac{ZL}{2}$$

$$M_A = -\frac{5549(64)^2}{2 \times 129} - \frac{8632 \times 129}{2} + \frac{22984 \times 129}{2} + \frac{4316 \times 129}{2}$$

$$M_A = 1115990$$

F_A is now:

$$F_A = \left(Z + S - \frac{\frac{N_1 l}{2}}{L} - N^1\right) = 4316 + 22984 - 2767 - 8632$$

$$= 15901 \text{ pounds.}$$

Moment $F_A = 15901 \times 129 = 2051229$ inch pounds
and F_B

$$F_B = \left(-Z + S - \frac{\frac{N_1 l}{2}}{L}\right) = -4316 + 22984 - 2767 = 15901 \text{ pounds.}$$

Moment $F_B = 15901 \times 129 = 2051229$ inch pounds.
Moment $(Z + S - N^1)L = 2408172$ inch pounds.

$$(-Z + S)L = 2408172 \text{ inch pounds.}$$

$$\left(\frac{N_1}{2}\right)l = 355136 \text{ inch pounds.}$$

$$N^1 L = 1113528 \text{ inch pounds.}$$

$$(Z + S)L = 3521700 \text{ inch pounds.}$$

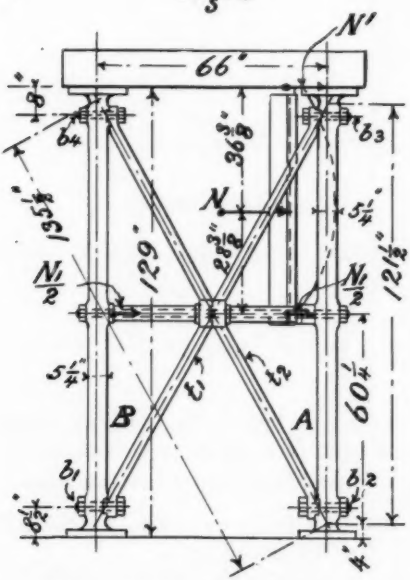
$$F_A + F_B = 2S - \frac{N_1 l}{L} - N^1 = 31802 \text{ pounds.}$$

Taking the frames of Fig. 3, as they are designed, with the tie rods to take up the bending stresses due to the forces S and N, we find the tie rods t_1 and t_2 (see Fig. 8), having a section of $(2\frac{5}{8}$ inches$)^2 = 6.89$ square inches.

The load to be taken up is again the load produced by the engine in ahead motion and in the maximum rolling position in direction of the force N, that is with maximum roll towards the guides.

The load is $F_A + F_B = 2S + \frac{N_1 l}{L} + N^1$ and $= 60{,}106$ pounds and is to be cared for by the two tie rods t_1.

Fig. 8



From the parallelogram of forces we obtain (see Fig. 8),

$$66 : 135.5 = 60106 : t_1,$$

$$123395 \text{ pounds} = t_1$$

$$\frac{123395}{2 \times 6.89} = 8955 \text{ pounds per square inch}$$

of section of the lower part of tie rods t_1 in tension. The upper half of the tie rods t_2 take a share of the force $N_1 = 11098$ pounds, this share is:

$$33 : 69.5 = 11098 : t_2 \text{ top end},$$

$$23377 = t_2 \text{ top end},$$

$$\frac{23377}{2 \times 6.89} = 1696.1 \text{ pounds per square inch.}$$

The upper part of t_1 is relieved by one-half of this amount, therefore, the tensile stress at the top end of the tie rods t_1 is only

$$8955 - 848 = 8107 \text{ pounds per square inch.}$$

Each of the two $2\frac{1}{4}$-inch connecting bolts t_3* are under 8 tensile stress, due to the arrangement of the tie rods, as shown in Fig. 8, of

$$\frac{60106}{2 \times 3.976} = 7558.6 \text{ pounds per square inch.}$$

The $2\frac{1}{2}$-inch bolts, b_1, are each under a tensile stress of:

$$\frac{60106}{2 \times 4.9} = 6133 \text{ pounds per square inch.}$$

The $2\frac{1}{2}$-inch bolts, b_3, have to withstand a tensile stress of:

$$\frac{60106 - 11098}{2 \times 4.9} = 5001 \text{ pounds per square inch,}$$

and the bolts, b_4, only

$$\frac{11098}{2 \times 4.9} = 1132.4 \text{ pounds per square inch.}$$

None of these bolts are under a shearing stress, as is shown in Fig. 8.

The connection of the tie rods t_1 produces a bending moment upon column B parallel to N of:

$$60106 \times 64 \times \left[\left(1 - \frac{8.5}{64}\right)^2 - \left(1 - \frac{8.5}{64}\right)^3\right] = 384678 \text{ inch pounds,}$$

and the component of the stress on the tie rods t_1 a tensile stress upon the lower end of column B and a compressive stress on the lower end of housing A of:

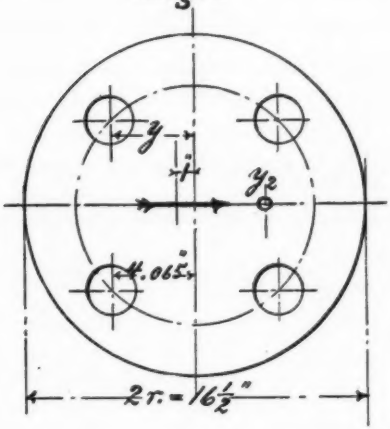
$$66 : 121.5 = 60106 : x$$

$$x = 110646 \text{ pounds,}$$

$$\frac{110646}{8} = 13831 \text{ pounds on each of the lower bolts of column } B.$$

* Bolts t_3 are the bolts which connect housing A and column B at the lower support of the guide; see (Fig. 8).

Fig. 9



The two outermost bolts of each of the lower flanges B, (see Figs. 8 and 9), are subject to a tensile stress due to the bending moment of:

$$\frac{384678}{2} = 192339 \text{ inch pounds} = M;$$

the equation for M is:

$$M = 2s_1y_1 + 2s_1y_2$$

$$y_2 = \frac{\frac{1}{2} \text{ moment of Inertia}}{\frac{1}{2} \text{ statical moment}}$$

of the disc area of the flange.

$$y_2 = \frac{\frac{\pi}{128} \times D^4}{\frac{\pi}{8} D^2 \times \frac{2}{3\pi} D} = \frac{3\pi D}{32}.$$

y_2 is the distance of the point of resistance of the flange against M or against 192339 inch pounds, measured from the center line xx, and in the direction of N and S. If there was only one bolt, s^1 would be the stress on this bolt at a distance of one inch from the center line xx, and would be equal to the total moment = 192339 inch pounds.

Now as $s_1 = s^1 y_1$, therefore $s_1 y_1 = s^1 y^2$, and further $M = 2s^1 y_1^2 + py_2$, in which $p =$ moment at the point of resistance.

But at the distance y_2 is the point of reaction, consequently

$$p = 2s_1 = 2s^1 y_1$$

and

$$M = 2s^1 y_2^2 + (2s^1 y_1)\, y_2$$

and while $s^1 = \frac{s_1}{y_1}$ therefore we have

$$M = \frac{s_1}{y_1}(2y_1^2) + \frac{s_1}{y_1}(2y_1)\, y_2$$

and

$$s_1 = \frac{M}{2y_1 + 2y_2}.$$

$$s_1 = \frac{192339}{8.13 + 9.72} = 10775 \text{ pounds}$$

tensile stress on each of these outermost bolts B due to bending.

The $8 - 2\frac{1}{4}$-inch bolts at the lower end of column B are under a shearing stress of:

$$\frac{60106}{8} = 7513 \text{ pounds each,}$$

equal to $7513 \times \frac{5}{4} = 9391$ pounds tensile stress.

The admission pressure of the cylinder is 28 pounds absolute. The area of the piston of 78 inch diameter $= 4778.4$ square inch. Assuming 4 pounds back pressure, then we have:

$$4778.4 \times 24 = 114682 \text{ pounds.}$$

$$W = 51250 \times \cos. 40^\circ = \underline{38678} \text{ pounds.}$$

$$76004 \text{ pounds.}$$

$$\frac{76004}{16} = 4750 \text{ pounds}$$

tensile stress on each lower, and on each upper bolt of the frames, due to the steam pressure on the down stroke.

For each of the two lower outermost bolts on each of the two columns forming column B we have

13831 pounds tensile stress due to the tie rods t_1,
10775 pounds tensile stress from bending action,
4750 pounds tensile stress from pressure on piston,
9391 pounds tensile stress due to shearing stress.
38747 pounds.

$\frac{38747}{3.023} = 12817$ pounds per square inch of section at bottom of thread.

The two outermost bolts of each of the two flanges on the top end of housing A are subject to a tensile stress due to bending moment produced by the tie rods t_1, parallel to N, of

$$60106 \times 64\left[\left(1-\frac{8}{64}\right)^2-\left(1-\frac{8}{64}\right)^3\right] = 368128 \text{ inch pounds.}$$

The top flanges of 17 inches diameter have each $4-2$ inch bolts with a pitch circle of 13 inch. We have for y_1:

$$y_1 = \sin 45^\circ \times 6.5 = 4.5975 \text{ inch,}$$

and for y_2:

$$y_2 = \frac{3\pi \times 17}{32} = 5 \text{ inch}$$

further is

$$s_1 = \frac{184064}{9.2+10} = 9587 \text{ pounds for each of the outermost bolts of housing } A.$$

9587 pounds tensile stress from bending action,
4750 pounds tensile stress from pressure on piston,
9391 pounds tensile stress due to shearing stress.
23728 pounds.

$\frac{23728}{2.3} = 10317$ pounds per square inch of section at bottom of thread.

The columns B have each to stand a bending stress at the lower end of

$$\frac{192339 \text{ inch pounds}}{Z_1} = \frac{192339}{14.2} = 13545 \text{ pounds per square inch,}$$

in which Z_1 represents the modulus of section, and the housings A at the upper end equal to:

$$\frac{184064}{14.2} = 12962 \text{ pounds per square inch.}$$

Each of the single columns B have to withstand a tensile stress at the lower end, and each of the single housings A a compressive stress at the lower end of

$$\frac{110646}{2} = 55323 \text{ pounds due to the component of stress } t_1.$$

$$\frac{55323}{21.647} = 2556 \text{ pounds per square inch.}$$

Each of the single columns or housings has to stand a tensile stress due to the steam pressure of

$$\frac{76004}{4 \times 21.647} = 878 \text{ pounds per square inch.}$$

Each single column or housing has to withstand a shearing stress equal to a bending stress of

$$\frac{60106}{2 \times 21.647} \times \frac{5}{4} = 1388 \text{ pounds per square inch.}$$

These latter stresses act at the top ends of housings A and at the bottom ends of columns B.

Total at top end of housings A $= 15228$ lbs. per square inch.
Total at bottom end of columns $B = 18367$ lbs. per square inch.

In my opinion no tie rod should be treated as being in compression. In this arrangement of the tie rods, which is a very rigid connection, there may have been allowed a compressive stress on the tie rods. This would reduce the calculated stress on columns and bolts to somewhat half the amount.

The bending moments due to the tie-rod connection produce a deflection, or rather a double deflection, of the column. For example, take tie rod t_1 (Fig. 8). The stress from it on the lower end of column B would tend to bend column B above the b_1 towards the center of the engine, while the stress from it on

upper end of housing A would have the same effect on housing A below b_3, but the horizontal connection between B and A will maintain the columns in position and the effect of the deflection, which is at its greatest but very small, will not be noticeable on the quiet working of the engine.

If the engine reversed its motion, and the ship is rolling in the same direction as the motion of the engine, the stress on the frames will be exactly the same as in ahead motion, because the force Z disappears, as can be seen from the sum of

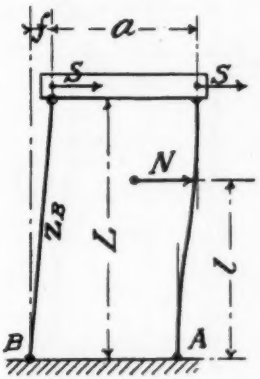
$$F_A + F_B = 2S + \frac{N_1 l}{L} + N^1.$$

Finally, in this example there is another point to cover. As can be seen in Fig. 3, the cylinder is still stable at a rolling angle of 40 degrees, therefore there cannot be produced any additional tensile stress on the frames by the rolling motion; in some cases it appears that the cylinder becomes unstable, but to such a small extent that it will not have much effect. This can be prevented under all circumstances by increasing the distance between the frames in the athwartship plane. It is not necessary to have the columns B and housing A located at equal distances from the center line of the cylinder in the athwartship plane.

In engines with radial valve gears, where there is always a great overhang on one side due to the valve chest, this side being opposite to the ahead guide side, the housing A can always be located at its proper distance in order to avoid too much weight in the crosshead and guide slipper. The column B can be located so that the weight of the cylinder is carried more equally by both frames B and A. The resulting unequal distribution of the load due to the steam pressure upon the bottom of the cylinder can be regulated by properly designed and located ribs; the stress on the tie rods is thereby very much reduced.

If column B is of such a form and is so connected to the cylinder as to allow of considering it as being hinged at each end, while housing A is fastened to the bedplate and to the cylinder as described before, we have, as shown in Fig. 10,

Fig. 10



$$\frac{1}{J_A E_A}\left(\frac{Nl^2}{2}+\frac{2SL^2}{2}-Z_B a L\right)=\frac{dy}{dx}$$

$$Z_B=\frac{Nl^2+2SL^2}{2aL}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The deflection is, in this case:

$$f=\frac{1}{J_A E_A}\left(\frac{Nl^3}{3}+\frac{Nl^2}{2}(L-l)+\frac{2SL^3}{3}-\frac{Z_B a L^2}{2}\right). \quad . \quad (6)$$

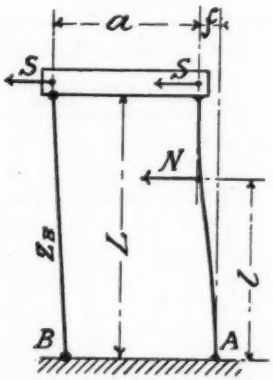
or if the value of Z_B from equation (5) is placed in equation (6)

$$f=\frac{1}{J_A E_A}\left[N\left(\frac{l^2 L}{4}-\frac{l^3}{6}\right)+\frac{SL^3}{6}\right]. \quad . \quad . \quad . \quad (6a)$$

If the engines reverse their motion and the ship is rolling in the opposite direction, the equations (5, 6 and 6a) will give the same result, but the stress Z_B is now a compressive stress on column B, as can be seen in Fig. 10a.

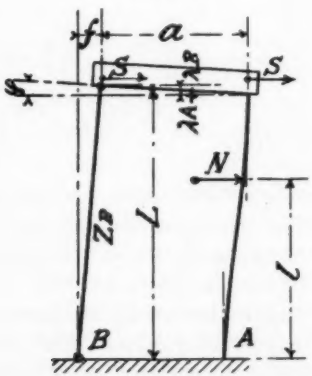
Of course this supposition would be incorrect if it was not generally agreed that forged steel and steel castings have their tensile and compressive strengths of the same value. The stress

Fig. 10a



Z_B tends to decrease the length of the housing A, but to increase that of column B, while the engine is in ahead motion and rolling in direction of the force N. If $\mathfrak{A}_A$ and $\mathfrak{A}_B$ represent the respective sections of the frames, the amount of decrease (see Fig. 11) of A is equal to λ_A;

Fig. 11



$$\lambda_A = \frac{Z_B L}{\mathfrak{B}_A E_A}$$

and the increase of B is equal to λ_B,

$$\lambda_B = \frac{Z_B L}{\mathfrak{B}_B E_B}.$$

The assumed crossbar in this case has not now a horizontal, but an inclined position, with an angle of inclination of $\varphi = \frac{\lambda_A + \lambda_B}{a}$.

With regard to (Fig. 11) we have now

$$\frac{1}{J_A E_A}\left(+\frac{Nl^2}{2} + \frac{2SL^2}{2} - Z_B \gamma L\right) = \varphi = \frac{Z_B L}{a}\left(\frac{1}{\mathfrak{B}_A E_A} + \frac{1}{\mathfrak{B}_B E_B}\right),$$

and from this equation

$$Z_B = \frac{a(Nl^2 + 2SL^2) \cdot \mathfrak{B}_A \mathfrak{B}_B E_B}{2L[a^2 \mathfrak{B}_A \mathfrak{B}_B E_B + J_A(\mathfrak{B}_A E_A + \mathfrak{B}_B E_B)]} \quad . \quad (7)$$

For example, we take the frames of the former example with a rectangular section of 24-inch × 18-inch × $1\frac{3}{8}$-inch = 324 square inches = $\mathfrak{B}_A$ and 25,000,000 = E_A.

The two front columns, which represent column B, should have for each $J_B = 3730$ inch^4, for $E_A = 30{,}000{,}000$. Their sections will be 20-inch diameter and $17\frac{1}{4}$-inch diameter of hole. The section is then 80.5 square inches.

We assume the guide faces to be supported over their entire length, then l (see Fig. 8) is 88.5 inches, and N, as before, 19730 pounds. $a = 66$ inch.

According to equation 7, we have for Z

$$Z_B =$$

$$\frac{66 \times [2 \times 22984 \times (129)^2 + 19730 \times (88.5)^2] \times 324 \times 161 \times 30{,}000{,}000}{129[(66)^2 \times 324 \times 161 \times 25{,}000{,}000 + 8952(324 \times 30{,}000{,}000 + 161 \times 25{,}000{,}000)]}$$

$$Z_B = 100590 \text{ pounds.}$$

The deflection f in this case, from equation (6a), is

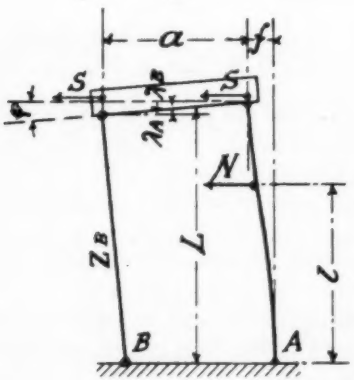
$$f = \frac{1}{8952 \times 25{,}000{,}000}$$
$$\left[19730\left(\frac{(88.5)^2 \times 129}{4} - \frac{(88.5)^3}{6}\right) + \frac{22984 \times (129)^3}{6}\right]$$
$$f = 0.528 \text{ inch.}$$

The quantities of λ_A and λ_B are very small, their sum being only

$$\lambda_A + \lambda_B = 100590 \times 129 \times \left[\frac{1}{324 \times 25{,}000{,}000} + \frac{1}{161 \times 30{,}000{,}000}\right]$$
$$\lambda_A + \lambda_B = 0.0036 \text{ inch,}$$

but the deflection is very remarkable and shows how necessary a proper staying of marine engines is to overcome the strains due to rolling.

Fig. 11a



If we reverse the conditions again, as shown in Fig. 11a, we have the same amount of Z_B and of the deflection, with the original assumption of column B and housing A, that is $J_B E_B$ equal to $J_A E_A$ and the equation (6) and (7) will give the amount. It is clear, however, as shown in Fig. 11a, that Z_B is again a compressive stress, column B will be shortened and housing A lengthened by the amount λ_B and λ_A respectively and the deflection will also be in the opposite direction. Equations (6) and

(7) will answer correctly also for the above example, where there is a difference between $J_A E_A$ and $J_B E_B$, but with the tensile strength of the material equal to the compressive.

If the frames are inclined to each other, which is generally the case if both frames are castings or housings, as shown in

Fig. 12

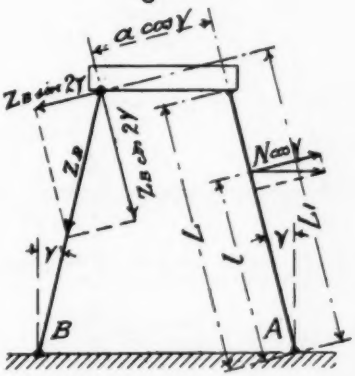


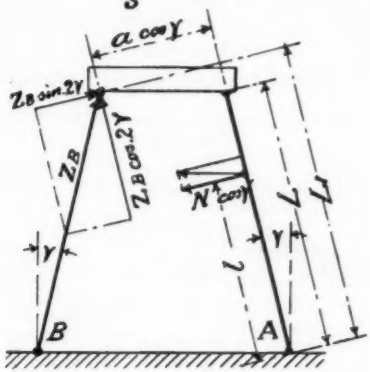
Fig. 12, the force N will act through its components upon the frames, with $N \cdot \cos \gamma$ in the normal direction and with $N \cdot \sin \gamma$ in the direction of the axis A. The force S on the upper end of column B, as well as the force S on the upper end of housing A, forces not shown in Figure, are both acting with their components, in the same direction as the force N, because the engine is again in ahead motion and rolling towards N.

Z_B will be reduced into $Z_B \cdot \sin 2\gamma$, normal to A, and $Z_B \cdot \cos 2\gamma$ parallel to A.

In Fig. 12a is shown the direction of N and Z_B in the reversed condition. The forces S, which are not shown, act again at the top end of the frames A and B in direction parallel with the force N and the components of N.

In the first case, by bending, the upper end of A is lifted, through the distance Δ_A, but at the same instant the upper end of B is lowered through the distance Δ_B.

Fig. 12a

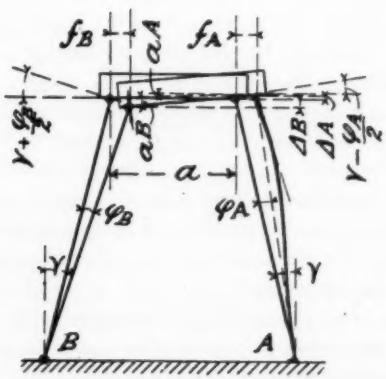


The amount of the angles aA and aB is given by

$$\alpha_A = \frac{\Delta_A}{a}; \quad \alpha_B = \frac{\Delta_B}{a}$$

see Fig. 13.

Fig. 13



From which we derive

$$\Delta_A = \frac{f_A \sin\left(\gamma - \frac{\varphi_A}{2}\right)}{\cos\left(\gamma - \frac{\varphi_A}{2}\right)}$$

and

$$\Delta_B = \frac{f_B \sin\left(\gamma + \frac{\varphi_B}{2}\right)}{\cos\left(\gamma + \frac{\varphi_B}{2}\right)}.$$

These values, due to the supposed deflection, must be very small, and the angles

$$\frac{\varphi_A}{2} = \frac{f_A}{2L\left(\cos\gamma - \frac{\varphi_A}{2}\right)} \text{ and } \frac{\varphi_B}{2} = \frac{f_B}{2L\left(\cos\gamma + \frac{\varphi_B}{2}\right)}$$

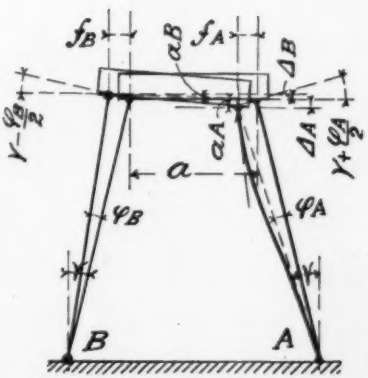
can be neglected in comparison with the angle γ, and we have, if

$$f_A \text{ assumed} \overset{\infty}{=} f_B$$
$$\Delta_A + \Delta_B \overset{\infty}{=} 2f.\tan\gamma,$$

and further

$$a_A + a_B \overset{\infty}{=} \frac{2f\tan\gamma}{a}.$$

Fig. 14



In the second case, by bending, see Fig. 14, the upper end of A is lowered through the distance Δ_B and at the same time the upper end of B is raised through the distance Δ_B.

The amount of the angles is again given by

$$a_A = \frac{\Delta_A}{a};\ a_B = \frac{\Delta_B}{a}$$

and

$$\Delta_A = \frac{f_A \sin\left(\gamma + \frac{\varphi_A}{2}\right)}{\cos\left(\gamma + \frac{\varphi_A}{2}\right)},$$

$$\Delta_B = \frac{f_B \sin\left(\gamma - \frac{\varphi_B}{2}\right)}{\cos\left(\gamma - \frac{\varphi_B}{2}\right)}$$

For the same reasons as before, the angles

$$\frac{\varphi_A}{2} = \frac{f_A}{2L\left(\cos\gamma + \frac{\varphi_A}{2}\right)} \text{ and } \frac{\varphi_B}{2} = \frac{f_B}{2L\left(\cos\gamma - \frac{\varphi_B}{2}\right)}$$

can be neglected in comparison with the angle γ, and we have again, if

$$f_A \text{ assumed} \overset{\infty}{=} f_B$$

$$\Delta_A + \Delta_B \overset{\infty}{=} 2f \cdot \tan\gamma,$$

and

$$a_A + a_B \overset{\infty}{=} \frac{2f \tan\gamma}{a}.$$

If the changes of lengths of A and B, by the applied force, are not taken into consideration, on account of their very small values as shown by the value of $\lambda_A + \lambda_B$ of the last example, we have, from the former assumption and with reference to Figs. 12 to 14, the following equations, which answer for both cases:

$$\frac{1}{J_A E_A}\times$$

$$\left[\frac{N\cos\gamma l^2}{2}+\frac{2S\cos\gamma L^2}{2}-Z_B\left(La\cos\gamma\cos 2\gamma+\left(L_1L-\frac{L^2}{2}\right)\sin 2\gamma\right)\right]$$

$$+\frac{2f\tan\gamma}{a}=0$$

$$\frac{2f\tan\gamma}{a}=\frac{1}{J_A E_A}\times$$

$$\left[Z_B\left(La\cos\gamma\cos 2\gamma+\left(L_1L-\frac{L^2}{2}\right)\sin 2\gamma\right)-\frac{N\cos\gamma l^2}{2}-\frac{2S\cos\gamma l^2}{2}\right]$$

therefore the deflection

$$f=\frac{a}{2J_A E_A\tan\gamma}\left[Z_B\left(La\cos\gamma\cos 2\gamma+\left(L_1L-\frac{L^2}{2}\right)\sin 2\gamma\right)\right.$$

$$\left.-\frac{N\cos\gamma l^2}{2}-\frac{2S\cos\gamma L^2}{2}\right]. \quad . \quad . \quad . \quad (8)$$

But the deflection f is also derived from

$$\frac{dy}{dx}=\frac{df}{dL}=\frac{1}{J_A E_A}\left[N\cos\gamma\frac{l^2}{2}\right.$$

$$\left.+\frac{2S\cos\gamma L^2}{2}-Z_B\left(La\cos\gamma\cos 2\gamma+\left(L_1L-\frac{L^2}{2}\right)\sin 2\gamma\right)\right]$$

$$f=\frac{1}{J_A E_A}\left[N\cos\gamma\left(\frac{Ll^2}{2}-\frac{l^3}{6}\right)\right.$$

$$+\frac{2S\cos\gamma L^3}{6}-Z_B\left(\frac{aL^2}{2}\cos\gamma\cos 2\gamma\right.$$

$$\left.\left.+\left(\frac{L_1L^2}{2}-\frac{L^3}{6}\right)\sin 2\gamma\right)\right]. \quad . \quad . \quad . \quad (8a)$$

From equation 8 and 8a, both being equal, we obtain the value of Z_B by forming a new equation, as follows:

$$Z_B=$$

$$\frac{N\cos\gamma\left(\frac{l^2a}{2\tan\gamma}+Ll^2-\frac{l^3}{3}\right)+2S\cos\gamma\left(\frac{L^2a}{2\tan\gamma}+\frac{L^3}{3}\right)}{L\left[\frac{a^2\cos\gamma\cos 2\gamma}{\tan\gamma}+\left(L_1-\frac{L}{2}\right)\frac{a\sin\gamma}{\tan\gamma}+La\cos\gamma\cos 2\gamma+\left(L_1L-\frac{L^2}{3}\right)\sin 2\gamma\right]}.$$

. . . 9

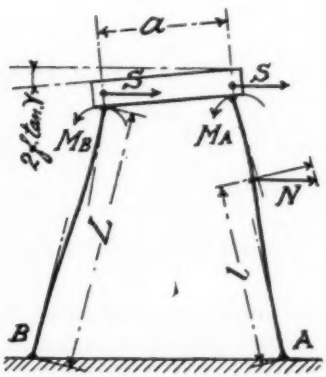
If we introduce again the two signs of + and — in the equations in order to obtain the stresses on the frames at the two extreme rolling positions, while the engine is in ahead motion, equation 9 will read:

$$Z_B = \frac{\mp N\cos\gamma\left(\frac{l^2 a}{2\tan\gamma}+Ll^2-\frac{l^3}{3}\right)+2S\cos\gamma\left(\frac{L^2 a}{2\tan\gamma}+\frac{L^3}{3}\right)}{L\left[\frac{a^2\cos\gamma\cos 2\gamma}{\tan\gamma}+\left(L_1-\frac{L}{2}\right)\frac{a\sin\gamma}{\tan\gamma}+La\cos\gamma\cos 2\gamma+\left(L_1 L-\frac{L^2}{3}\sin 2\gamma\right)\right]} \quad . \quad . \quad . \quad (9)$$

The lower sign of + and — has to be used in the following equations until further notice.

Returning now to the original assumption for column B, that it is rigidly fastened to the bedplate and at the upper end, there is found further by the same reasoning from which we obtained equations 8 to 9, and with assistance to Fig. 15:

Fig 15



$$\frac{(\mp Z+S)\cos\gamma\frac{L^2}{2}-M_B L+2f\frac{\tan\gamma}{a}}{J_B E_B}=0$$

and

$$f=\frac{1}{J_B E_B}\left[M_B L-(\mp Z+S)\cos\gamma\frac{L^2}{2}+2f\frac{\tan\gamma}{2}\right];$$

but f can be also obtained by integrating the equation

$$\left[(\mp Z + S)\cos\gamma\frac{L^2}{2} - M_B L\right]\frac{1}{J_B E_B} = \frac{dy}{dx} = \frac{df}{dL}$$

which give

$$f = \frac{1}{J_B E_B}\left[\left((\mp Z + S)\cos\gamma\frac{L^3}{3}\right) - M_B\frac{L^2}{2}\right].$$

From these two equations of f we obtain

$$M_B = \frac{(\mp Z + S)\cos\gamma\left(\frac{2}{3}L^2 + \frac{La}{2\tan\gamma}\right)}{L + \frac{a}{\tan\gamma}} \quad . \quad . \quad . \quad (10)$$

By inserting the value of M_B in the last equation for f, we have:

$$f = \frac{1}{J_B E_B}(\mp Z + S)\cos\gamma\left(\frac{L^3}{3} - \frac{\frac{L^4}{3} + \frac{L^3 a}{4\tan\gamma}}{L + \frac{a}{\tan\gamma}}\right). \quad . \quad (11)$$

From equation

$$\frac{1}{J_A E_A}\left(\mp\frac{Nl^2}{2} + \frac{SL^2}{2} \pm \frac{ZL^2}{2} - M_A L\right) = 0$$

and by the former reasoning we have for housing A:

$$\frac{1}{J_A E_A}\left[\mp\frac{N\cos\gamma l^2}{2} + \frac{S\cos\gamma L^2}{2} - M_A L \pm Z\cos\gamma\frac{L^2}{2} + \frac{2f\tan\gamma}{a}\right] = 0$$

$$f = \frac{1}{J_A E_A}\left[\left(\mp\frac{Z\cos\gamma L^2}{2} + M_A L \pm N\cos\gamma\frac{l^2}{2}\right)\frac{a}{2\tan\gamma} - \left(S\cos\gamma\frac{L^2}{2}\right)\frac{a}{2\tan\gamma}\right]. \quad . \quad (12)$$

The second equation for the deflection f of housing A we obtained by integrating the equation:

$$\frac{1}{J_A E_A}\left(\mp\frac{N\cos\gamma l^2}{2} + \frac{S\cos\gamma L^2}{2} - M_A L \pm Z\cos\gamma\frac{L^2}{2}\right) = \frac{dy}{dx} = \frac{df}{dL}$$

and by integration we have

$$f=\frac{1}{J_A E_A}\left(\mp N\cos\gamma\left(\frac{Ll^2}{2}-\frac{l^3}{6}\right)+\frac{S\cos\gamma L^3}{3}-M_A\frac{L^2}{2}\pm Z\cos\gamma\frac{L^3}{3}\right). \quad . \quad (12a)$$

and from the combination of equations (12) and (12a) is found the value of

$$M_A=$$
$$\frac{\mp N\cos\gamma\left(l^2+\frac{l^2a}{2L\tan\gamma}-\frac{l^3}{3L}\right)+S\cos\gamma\left(\frac{2}{3}L^2+\frac{La}{2\tan\gamma}\right)\pm Z\cos\gamma\left(\frac{2}{3}L^2+\frac{La}{2\tan\gamma}\right)}{L+\frac{a}{\tan\gamma}}.$$

. . . (13)

The obtained value of M_A placed in equation (12a) give for

$$f=\frac{1}{J_A E_A}\left[\mp N\cos\gamma\left(\frac{Ll^2}{2}-\frac{l^3}{6}-\frac{\frac{L^2l^2}{2}+\frac{Lal^2}{4\tan\gamma}-\frac{Ll^2}{6}}{L+\frac{a}{\tan\gamma}}\right)+\right.$$
$$\left.S\cos\gamma\left(\frac{L^3}{3}-\frac{\frac{L^4}{3}+\frac{L^3a}{4\tan\gamma}}{L+\frac{a}{\tan\gamma}}\right)\pm Z\cos\gamma\left(\frac{L^3}{3}-\frac{\frac{L^4}{3}+\frac{L^3a}{4\tan\gamma}}{L+\frac{a}{\tan\gamma}}\right)\right]$$

. . . (14)

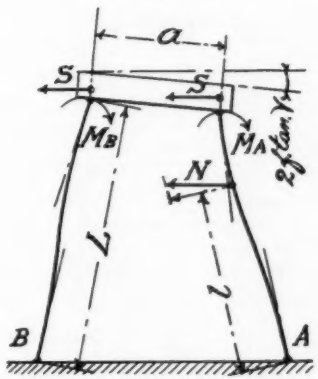
and from this result, finally, by combination of equations (11) and (14), we derive Z.

$$Z=$$
$$\frac{\left(N\left[\frac{Ll^2}{2}-\frac{l^3}{6}-\frac{\frac{L^2l^2}{2}+\frac{Lal^2}{4\tan\gamma}-\frac{Ll^2}{6}}{L+\frac{a}{\tan\gamma}}\right]J_BE_B\right)+S\left(\frac{L^3}{3}-\frac{\frac{L^4}{3}+\frac{L^3a}{4\tan\gamma}}{L+\frac{a}{\tan\gamma}}\right)(\mp J_BE_B\pm J_AE_A)}{\left(\frac{L^3}{3}-\frac{\frac{L^4}{3}+\frac{L^3a}{4\tan\gamma}}{L+\frac{a}{\tan\gamma}}\right)(J_AE_A+J_BE_B).}$$

. . . (15)

From equation (15) it is again seen that the force S has no influence upon the force Z, as long as the column B and the housings A are of the same size and material. as stated before.

Fig. 16



By comparison of Fig. 15 and Fig. 16, we can see that the equations (10) to (15) answer also for both extreme rolling cases.

For example, thus:

The low-pressure cylinder of a triple-expansion engine (Fig. 17), has an actual area of 4417.9 square inches. Admission pressure, 28 pounds absolute.

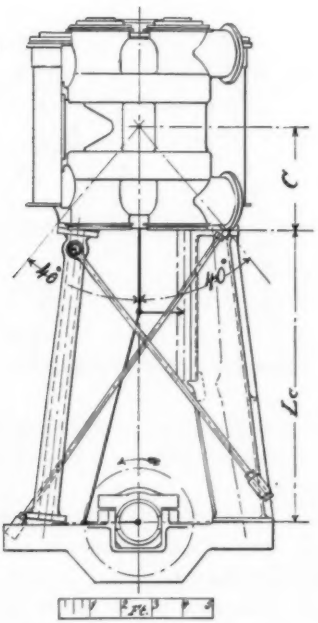
The cylinder is connected to the bedplate by two hollow forged-steel columns, forming the column B.

These columns are provided with flanges at the top and bottom ends. The diameter of the columns at the top end, near the fillet of the flange, is 7 inches, and the diameter at the bottom end, 9½ inches; the axial hole is 5 inches diameter. The housing A, which forms the connection between the cylinder and bedplate at the opposite side in the athwartship plan is of cast steel with a shape as shown in Fig. 17, the view of it in a fore-and-aft elevation forming a reversed y (ʎ).

The guides are supported over their entire lengths by the

housing *A*. The arrangement and connection of the two pairs of rolling ties, forming two diagonal crosses in the athwartship plan, is shown in Fig. 17.

Fig. 17



The inclination of column *B* is 12.75 inches, while the inclination of the neutral line of the steel casting of housing *A* is only 11.54 inches; but by adding the cast-iron guide the inclination changes to 16.5 inches, though the difference in strength of these two materials will reduce the inclination remarkably; therefore, and for simplifying the equations, I have assumed, without great error, that the inclination of both frames being equal, and 12.75 inches on a length *L* of 113.2 inches,

$\text{Cos } \gamma L = L_c = 112.5$ inches.
Angle $\gamma = 6$ degrees, 28 minutes, 30 seconds.
$W = 43000$ pounds.
$W_1 = 58283$ pounds.
$a = 52.5$ inches.
$C = 40$ inches.
$S = 18733$ pounds.
$N = 16739$ pounds.
$l = 84$ inches.
$J_A\,(\text{mean}) = 4100$ inch^4.
$J_B\,(\text{mean}) = 415$ inch^4.
$E_A = 25{,}000{,}000$.
$E_B = 30{,}000{,}000$.

Neglecting at first the rolling ties, for the same reason as in the first example, and assuming the engine in ahead motion, rolling 40 degrees in the direction of N, we obtain from equation (15), by using the lower signs of + and — for Z:

$$a = \frac{113.2\times(84)^2}{2} - \frac{(84)^3}{6}$$

$$b = \frac{\dfrac{(113.2)^2\times(84)^2}{2} + \dfrac{113.2\times 52.5\times(84)^2}{4\times 0.1135} - \dfrac{113.2\times(84)^3}{6}}{113.2 + \dfrac{52.5}{0.1135}}$$

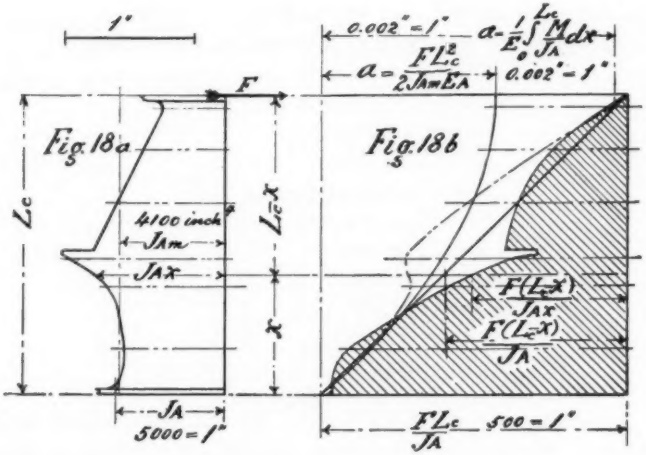
$$c = 16739\times 12{,}450{,}000{,}000$$

$$d = 18733\times 109275\times(-9{,}005{,}000{,}000)$$

$$e = \left(\frac{(113.2)^3}{3} - \frac{\dfrac{(113.2)^4}{3} + \dfrac{(113.2)^3\times 52.5}{4\times 0.1135}}{113.2 + \dfrac{52.5}{0.1135}}\right)\times 114{,}950{,}000{,}000$$

$$Z = [\{[a-(b)]\times c\} + \{d\}] \div e.$$

$$Z = -13300 \text{ pounds.}$$



The moments of inertia of housing A from the top end down at sections, as indicated in Fig. 18a, but with the flanges at top and bottom ends omitted, are as following:

2400	inch⁴	6066.7	inch⁴
2400.24	"	4563.78	"
3112	"	4184.46	"
4205	"		

Further we obtain from equation 13 for M_A

$$M_A = \left[\begin{array}{l} \left[16739 \times 0.99362 \left(84^2 + \frac{(84)^2 \times 52.5}{2 \times 113.2 \times 0.1135} - \frac{(84)^3}{3 \times 113.2}\right)\right] \\ + \left[18733 \times 0.99362 \left(\frac{2}{3}(113.2)^2 + \frac{113.2 \times 52.5}{0.2270}\right)\right] \\ - \left[-13330 \times 0.99362 \left(\frac{2}{3}(113.2)^2 + \frac{113.2 \times 52.5}{0.2270}\right)\right] \end{array}\right] \div 113.2 + \frac{52.5}{0.1135}$$

$$M_A = 2494789 \text{ inch pounds.}$$

The force F_A contains the values

$$\left(-Z+S+\frac{Nl}{L}\right).$$

$F_A = -(-13330)+18733+12347 = 44410$ pounds.
Moment $= F_A L_c = 44410 \times 112.5 = 4996125$ inch pounds.

The value of the moment of F_A divided by the mean moment of inertia is shown by the triangle, which is marked at the base with

$$\frac{FL_c}{J_A}, \text{ see Fig. 18b.}$$

The value of the same moment divided by the moment of inertia of each indicated section, as shown by the cross-sectioned area af Fig. 18b, is for each indicated section from the top down:

0	430.1
83.3	700.7
292.6	1003
433	1194.1

A measure of the deflection of housing A at the top end with a mean J_A is given by the graphic method introduced by Mr. Mohr, which gives the full curved line with the value for the ordinate at the top of

$$a = \frac{FL^2c}{2J_A m E_A} = 0.002741,$$

and for a variable J_A is given, by a similar method, a measure with

$$a = \frac{1}{E}\int_0^{Lc} \frac{M}{J_A}\,dx$$

for the top ordinate of the curve shown in long and short dots, the value of a is:

$$a = 0.004685.$$

From equation (14) we obtain the deflection of housing A with a mean J_A to be:

$$f=\frac{1}{J_A E_A}\left[N\cos\gamma\left(\frac{Ll^2}{2}-\frac{l^3}{6}\right)-\left(\frac{\frac{L^2l^2}{2}+\frac{Lal^2}{4\tan\gamma}-\frac{Ll^3}{6}}{L+\frac{a}{\tan\gamma}}\right)\right.$$

$$\left.+S\cos\gamma\left(\frac{L^3}{3}-\frac{\frac{L^4}{3}+\frac{L^3a}{4\tan\gamma}}{L+\frac{a}{\tan\gamma}}\right)-Z\cos\gamma\left(\frac{L^3}{3}-\frac{\frac{L^4}{3}+\frac{L^3a}{4\tan\gamma}}{L+\frac{a}{\tan\gamma}}\right)\right]$$

$$f=\frac{(16632\times 81063)+(18613\times 109275)-(-13245\times 109275)}{102{,}500{,}000{,}000}$$

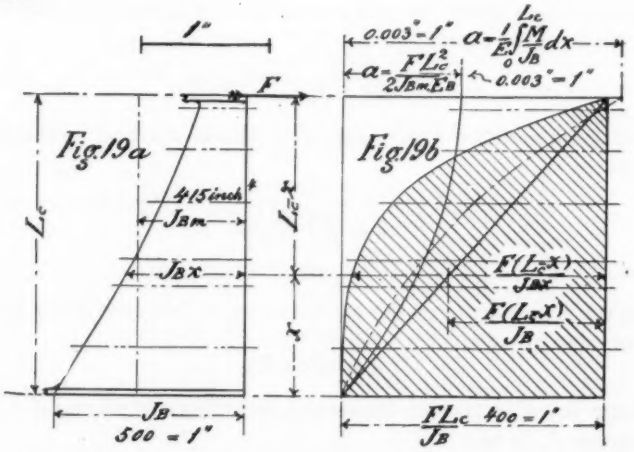
$f=0.047117$ inch.

Now we have a relative measure for the deflection in both cases, and, from the equation of deflection we have for $f=0.0471$ inch.

Using the two different values of ordinate (a)

$$0.002741 : 0.004685 = 0.0471 : f$$
$$0.0804 \text{ inch} = f_A.$$

This method of Mr. Mohr's is explained in the article of "The Stresses on Engine Frames of Stationary Engines," page 36 of this year's JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.



In Fig. 19a are shown the moments of inertia of the combined column B, also from the top down. Their amounts are as follows:

175 inch^4	426 inch^4
189 "	483 "
246 "	633 "
327 "	739 "

The moment M_B is given by equation 10:

$$M_B = \left[\frac{5043 \times 0.99362\left(\frac{2}{3} \times (113.2)^2 + \frac{113.2 \times 52.5}{2 \times 0.1135}\right)}{113.2 + \frac{52.5}{0.1135}}\right].$$

$$M_B = 324668 \text{ inch pounds.}$$

F_B contain the values $Z + S$.

$$F_B = -13330 + 18733 = 5403 \text{ pounds.}$$

and

Moment $= F_B \times L_C = 5403 \times 112.5 = 607838$ inch pounds.

The moment of F_B divided by the mean moment of inertia is shown by the triangle marked at the base thus: $\frac{FL_C}{J_B}$. $F_B \times L_C$ divided by the moment of inertia of each indicated section is here also shown by the cross-sectioned area (see Fig. 19b), the quotient for each indicated section from the top down being

0	780
128.6	805.4
450.2	819.4
677.5	822.51

The obtained curve would lead us to believe that the combined column B was very weak, but if, for example, we take J at half the length for one column, which is

$$((8.25)^4 - 5^4)\,0.0491 = 196.77 \text{ inch}^4$$

we have

$$\frac{196.77}{4.125} = 47.696 \times 2 = 95.392 = Z_1.$$

$$\frac{5403 \times \frac{112.5}{2}}{95.392} = 3186 \text{ pounds per square inch}$$

of section, though J decreases too much towards the top. The full curved lines gives, for the ordinate at the top, the value $a =$

$$\frac{FL_c}{2J_{Bm}E_B} = 0.002746$$

for the mean J_B; and for the variable J_B we find for the value of the top ordinate of the curve shown in long and short dots, the amount

$$a = \frac{1}{E}\int_0^{L_c}\frac{M}{J_B}dx = 0.00651.$$

From equation 11 we get the deflection of column B:

$$f = \frac{1}{J_B E_B}\left[(Z+S)\cos\gamma\left(\frac{L^3}{3} - \frac{\frac{L^4}{3} + \frac{L^3 a}{4\tan\gamma}}{L + \frac{a}{\tan\gamma}}\right)\right],$$

$$f = \frac{5403 \times 0.99362 \times 109275}{12{,}450{,}000{,}000}$$

$$f = 0.0471.$$

In the same way as before, we obtain the deflection of column B:

$$0.002746 : 0.00651 = 0.0471 : f$$
$$0.1117 = f_B$$
$$0.0804 = f_A$$
$$\text{or } 0.0313 \text{ inch}$$

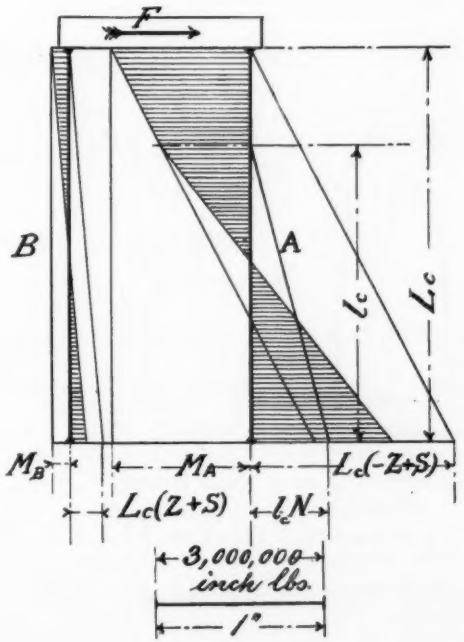
difference between deflection f_A and f_B

With a mean J_A and J_B, as used in the calculation, we found the deflection of housing A equal to that of column B, therefore this difference of the deflections is due to the more rapid decrease of J_B towards the top, compared with the decrease of J_A.

The great difference between J_B and J_A, and especially at the top ends of frames B and A, has an effect upon column B, as shown in Fig. 15. The angle ($2f\tan\gamma$)is increased, moving the top of column B closer towards the center of the engine. This is shown

already, by the value of Z, which is in this case $(-)$ minus. This implies that the stress in the cylinder bottom (or crossbar) is a compressive one; therefore the direction of the force Z is outwards. The force Z is the strain in the metal against compression. This compressive stress is produced by the force S on column B. Column B being much weaker than housing A, housing A has to withstand a part of S from column B, thereby forming the compressive stress in the cylinder bottom. This is exactly opposite to the conditions of the first example. The forces S had no effect on the crossbar, because J_B was equal to J_A, each frame taking care of its own equal portion of S and the force N only produced a tensile stress in the cylinder bottom, and therefore making the direction of Z inwards.

Fig. 20

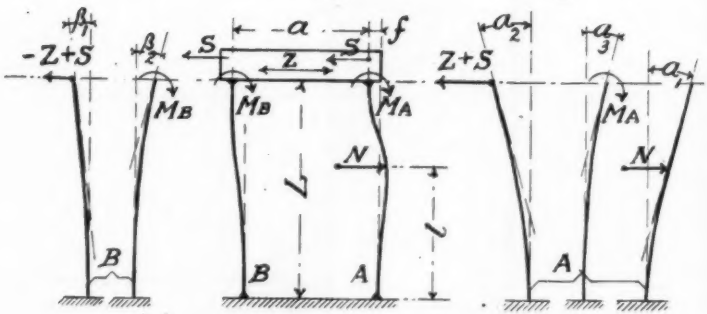


The increase of the deflection of column B beyond that of housing A would have an influence upon the quiet working of the engine. The bending moments as they appear over the entire length of the frames are shown in Fig. 20, and their values are as follows in inch pounds:

$$\begin{aligned} Lc(Z+S) &= 112.5\,(+(-13330)+18733) = 607838 \\ Lc(-Z+S) &= 112.5\,(-(-13330)+18733) = 3607087.5 \\ lcN &= 83.5\,(16739) = 1397706.5 \\ M_A &= 2494789 \\ M_B &= 324668 \end{aligned}$$

If the ship is now rolling 40 degrees in the opposite direction or away from the guide, but with the engine in ahead motion, the forces will act upon the frames as shown in Figs. 21, 21a and 21b, and we have to use again the upper signs of + and — in the equations (10), (11), (13), (14) and (15).

Fig. 21b Fig. 21 Fig. 21a



From equation (15) we obtain for Z:

$$Z = +\,16020 \text{ pounds.}$$

F_A now contains the values $\left(Z+S-\frac{Nl}{L}\right)$.

$$F_A = 16020 + 18733 - 12347 = 22406.$$

From equation (13) we have for M_A

$$M_A = \frac{-N\cos\gamma\left(l^2+\frac{l^2a}{2L\tan\gamma}-\frac{l^3}{3L}\right)+S\cos\gamma\left(\frac{2}{3}L^2+\frac{La}{2\tan\gamma}\right)+Z\cos\gamma\left(\frac{2}{3}L^2+\frac{La}{2\tan\gamma}\right)}{L+\frac{a}{\tan\gamma}}$$

$$M_A = \frac{-16632\times(7056+14416-1745)+18613\times(34813)+14918\times(34813)}{576}$$

$M_A = 1518271$ inch pounds.

Moment $F_A = \left(Z+S-\frac{Nl}{L}\right)\times Lc.$

Moment $F_A = 22406\times 112.5 = 2520675$ inch pounds.

F_B contains the values $(-Z+S)$.

$F_B = -16020+18733 = 2713$ pounds.

From equation (10) we obtain for M_B

$$M_B = \frac{2713\times 0.99362\left(\frac{2}{3}(113.2)^2+\frac{113.2\times 52.5}{2\times 0.1135}\right)}{113.2+\frac{52.5}{0.1135}}.$$

$M_B = 163025$ inch pounds.

Moment $F_B = (-Z+S)\times Lc.$

Moment $F_B = 2713\times 112.5 = 305212.5$ inch pounds.

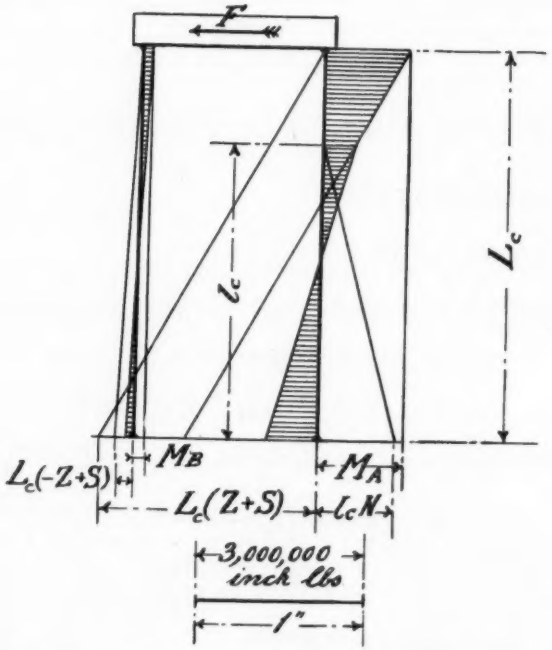
$$F_A+F_B = 2S-\frac{Nl}{L} = 25119 \text{ pounds.}$$

$(Z+S)L_C = 3909712.5$ inch pounds.

$Nl_C = 1397706.5$ inch pounds.

The bending moments as they now appear over the entire length of housing A and column B are shown in Fig. 22.

Fig. 22



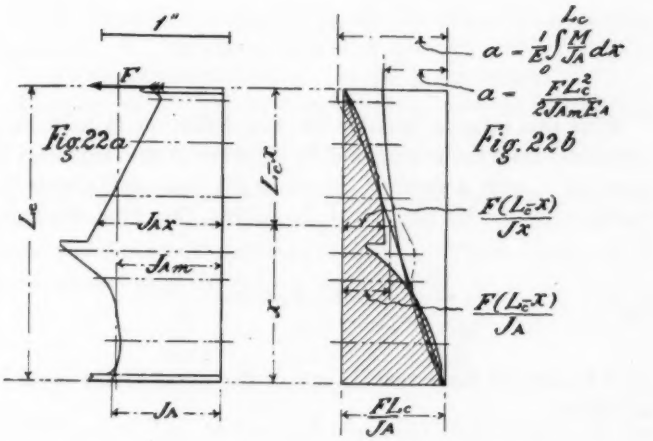
The deflection of housing A we obtain from equation 14, and is

$$f=\frac{1}{J_A E_A}\left[-N\cos\gamma\left(\frac{Ll^2}{2}-\frac{l^3}{6}-\left(\frac{\frac{L^2l^2}{2}+\frac{Lal^2}{4\tan\gamma}-\frac{Ll^3}{6}}{L+\frac{a}{\tan\gamma}}\right)\right.\right.$$

$$\left.\left.+S\cos\gamma\left(\frac{L^3}{3}-\frac{\frac{L^4}{3}+\frac{L^3a}{4\tan\gamma}}{L+\frac{a}{\tan\gamma}}\right)+Z\cos\gamma\left(\frac{L^3}{3}-\frac{\frac{L^4}{3}+\frac{L^3a}{4\tan\gamma}}{L+\frac{a}{\tan\gamma}}\right)\right],$$

$$f=\frac{(-16632\times 81063)+(18613.5\times 109275)+(15918\times 109275),}{102,500,000,000}$$

$$f=0.02366 \text{ inch.}$$



In Fig. 22a are shown the moments of inertia J of housing A at indicated sections, the flanges at top and bottom ends being omitted. The values are as already given. The values of the moments F_A divided by the mean moment of inertia are shown by the triangle (see Fig. 22b), which is marked at the base with $\frac{FL_c}{J_A}$.

The values of the same moment, divided by the moment of inertia of each indicated section, as shown by the cross sectioned area of Fig. 22b, are for each indicated section from the top end down:

0	234.6	596.3
45.11	243.7	647.03
158.6	379.7	

From the method introduced by Mr. Mohr we have the value of the top ordinate of the full curved line:

$$a = \frac{FL^2_c}{2J_{AM}E_A} = 0.001485,$$

and for a variable J_A the ordinate at the top of the curve, shown in long and short dots. The value of a is:

$$a = \frac{1}{E}\int_0^{L_c} \frac{M}{J_A} dx = 0.002581.$$

With this relative measure for the deflection in both cases again and from the corresponding equation of the deflection we have for f, with a mean J_A, $f = 0.02366$, and obtain with this measure the f with the actual J_A. Using, this time, the area y

$$y = \frac{1}{E_A}\int_0^{112.5} 0.002581 dx = \frac{16.333}{E_A}$$

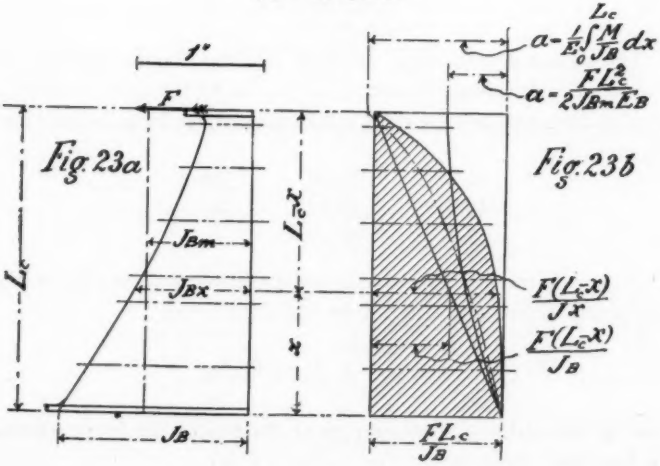
of the curve of the actual J_A and of the curve of the mean J_A, we obtain:

$$y = \frac{1}{E_A}\int_0^{112.5} 0.001485 dx = \frac{9.397}{E_A}.$$

then is:

$$9.397 : 16333 = 0.02366 : f$$

$$f_A = 0.04112.$$



In Fig. 23a are shown the moments of inertia of the combined column B.

The values of these moments are also given before.

The moments of F_B divided by the mean moment of inertia is shown by the triangle marked at the base with $\frac{FL_c}{J_B}$, see Fig. 23b.

The cross-sectioned area of Fig. 23b gives, with the boundary curve at each indicated section, the value of the moment F_B divided by the actual moment of inertia. These quotients of each indicated section from the top down are

0	340.1	411.44
64.6	391.7	413.1
226.1	404.4	

The full-curved line of Fig. 23b gives, for the ordinate at the top, the value a.

$$a = \frac{FL_c}{2J_{Bm}E_B} = 0.001379$$

for the mean J_B, but for the curved line in long and short dots we get the amount of the top ordinate =

$$a = \frac{1}{E}\int_0^{L_c} \frac{M}{J_B}\,dx = 0.003269.$$

The deflection of column B is found now from equation 11, with the upper signs of + and −

$$f = \frac{1}{J_B E_B}\left[(-Z+S)\cos\gamma\left(\frac{L^3}{3} - \frac{\frac{L^4}{3} + \frac{L^3 a}{4\tan\gamma}}{L + \frac{a}{\tan\gamma}}\right)\right],$$

$$f = \frac{2713 \times (0.99362 \times 109275)}{12{,}450{,}000{,}000}$$

$$f = 0.02366.$$

Using again the area y for comparison, we have from the curve of the actual J_B

$$y = \frac{1}{E_B} \int_0^{112.5} 0.003269\, dx = \frac{20.686}{E_B},$$

and from the curve of the mean J_B

$$y = \frac{1}{E_B} \int_0^{112.5} 0.001379\, dx = \frac{8.726}{E_B}$$

and the actual deflection

$$8.726 : 20.686 = 0.02366 : f$$
$$f_B = 0.05605.$$

Returning to the frames as they are designed, with the tie rods to take up the bending stress due to the forces S and N, we find the one pair of tie rods (see Fig. 17), which are connected with their top ends to lugs cast on the cylinder towards housing A and with their bottom end on the bedplate, these tie rods to be t_A. The opposite pair are connected each to the top end of one of the columns forming column B, while their lower ends are connected to lugs cast on each side of housing A, these tie rods to be t_B.

It can be seen, from Fig. 17, that when the engine is in ahead motion and rolling in the direction of force N, that there is no stress produced upon the housing A and the connecting bolts of housing A if the tie rods (t_A) are strong enough to take up the load

$$F_A + F_B = 2S + \frac{Nl}{L} = 49813 \text{ pounds.}$$

All tie rods are $1\frac{1}{2}$ inches diameter; therefore the actual area of one pair = 3.534 square inches. From the diagram of forces we obtain for 49813 pounds in the direction of t_A:

$$49813 : 79.75 = x : 137.9$$
$$86134.3 \text{ pounds} = x$$
$$\frac{86134.3}{3.534} = 24371 \text{ pounds per square inch of section.}$$

This shows that the tie rods are not designed for taking the total stress due to S and N. Assuming that the tie rods t_4 take $\frac{1}{2}$ of this amount per square inch of section and we reduce the load 49,813 pounds to 24,906 pounds. This acts on the tops of frames A and B in the direction of the force N, and produces a bending moment at the lower end of the frames of $24906 \times 112.5 = 2801925$ inch pounds. Now, from the investigation of these frames without roilling ties, we found

$$\begin{aligned} F_A \times L_c &= 4996125 \text{ inch pounds} \\ F_B \times L_c &= \underline{\ 607838 \text{ inch pounds}} \\ &\quad 5603963 \text{ inch pounds.} \end{aligned}$$

Without any great error we can assume that the reduced moment is taken up by housing A and column B proportionate to $F_A \times L_c$ and $F_B \times L_c$, and this moment for housing A will be

$$5603963 : 2801925 = 4996125 : x$$
$$x = 2498012 \text{ inch pounds.}$$

The moment M_A, which has the value of 2494789 inch pounds, is reduced by the tie rods in the same proportion.

The reduced $M_A = 1247370$ inch pounds.

The reduced value of $(F_A \times L_c - M_A) = 1251642$ inch pounds is the moment at the lower end of housing A above the flange.

The modulus of section at the lower end of housing A near the flange is $= 356.44 = Z_1$, then we have:

$$\frac{F_A \times L_c - M_A}{Z_1} = \frac{1250642}{356.44} = 3509 \text{ pounds tensile stress per sq. inch.}$$

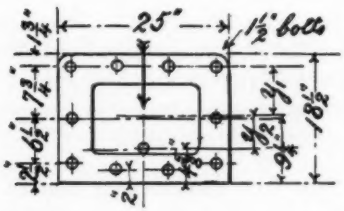
Z_1 for the top section is 356.91, very nearly the same as for the bottom section.

$$\frac{M_A}{Z_1} = \frac{1247370}{356.91} = 3545 \text{ pounds per square inch tensile stress at}$$

the top end of housing A.

The top flange is connected to the cylinder by ten $1\frac{1}{2}$-inch steel bolts, of which the active bolts have to withstand the moment 124730 inch pounds.

Fig. 24



According to Fig. 24, we have four 1½-inch bolts at a distance from the center line or the line xx. The second row of bolts are ½-inch $= y_{11}$ from the center line towards y_2. For the outermost bolts we have $s_1 = s^1 y_1$, and for the second row $s_2 = s^1 y_{11}$, therefore $s_1 y_1 = s^1 y_1^2$, and $s_2 y_{11} = s^1 y_{11}^2$, and further the moment

$$M = 4 s^1 y_1^2 + 2 s^1 y_{11}^2 + p y_2$$

p is again the moment at point of resistance, therefore $p = 4 s_1 + 2 s_2$ and the moment M also:

$$M = 4 s^1 y_1^2 + 2 s^1 y_{11}^2 + (4 s^1 y_1 + 2 s^1 y_{11}) y_2$$

and while $s^1 = \frac{s_1}{y_1}$, we have:

$$M = \frac{s_1}{y_1}(4 y_1^2 + 2 y_{11}^2) + \frac{s_1}{y_1}(4 y_1 + 2 y_{11}) y_2$$

and

$$s_1 = \frac{M}{\left(4 y_1 + \frac{2 y_{11}^2}{y_1}\right) + \left(4 + \frac{2 y_{11}}{y_1}\right) y_2}$$

$$M = M_A = 1247370 \text{ inch pounds.}$$

According to the method to find y_2 in the last example, we have the distance to the point of resistance 4⅜ inches from the center line, but it is perfectly safe to take the point of resistance in the center of the front wall of the housing A, which is 1½

inches from the inner edge of the top flange, this increases the distance y_2 to $7\frac{3}{4}$ inches, and the amount of s_1 will be:

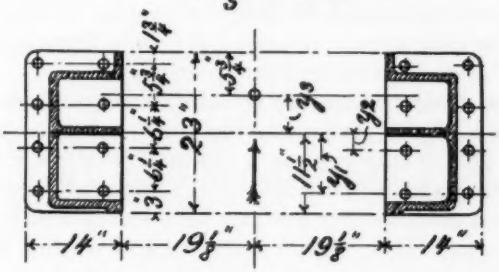
$$s_1 = \frac{1247370}{4\times 7.5 + \frac{2\times 0.0625}{7.5} + \left(4 + \frac{2\times 0.0625}{7.5}\right)\times 7.75}$$

$s_1 = 20449$ pounds for one bolt.

$\frac{20449}{1.294} = 15803$ pounds tensile stress per square inch of section of bolts at bottom of thread.

The bolts at the lower flanges are shown in Fig. 24a.

Fig. 24a



In each flange are eight $1\frac{3}{8}$-inch bolts, spaced as shown, and we obtain with the same formula as above for s_1:

$$s_1 = \frac{M}{\left(4y_1 + \frac{4y_2^2}{y_1}\right) + \left(4 + \frac{4y_2}{y_1}\right)y_3} \qquad M = (F_A \times L_C) - M_A.$$

Here we can also take the point of resistance of the bottom flange with safety in the middle of the outside wall; this would bring this point to within $3\frac{1}{2}$ inches from the outer edge of the flange.

This increases the distance y_3 to 8 inches, and the amount of s_1 will be:

$$s_1 = \frac{1250642}{34 + 2.35 + 40.464}$$

$s_1 = 16282$ pounds for one bolt.

$\frac{16282}{1.057} = 15404$ pounds tensile stress per square inch of section of bolt at bottom of thread.

The tie rods should have been $2\frac{3}{8}$ inches actual diameter, with an aggregate area for the two of 8.86 square inches. They would have taken up the whole stress due to the forces $F_A + F_B = 49{,}813$ pounds, which produce a stress on the tie rods of 86,134.3 pounds, as shown before.

$\frac{86134.3}{8.86} = 9722$ pounds per square inch of section of tie rods.

If the tie rods had taken the strain as indicated this would have relieved the bolts in the flanges of housing A and column B from all bending stress, and these bolts would have had to take only the stresses due to the steam pressure.

The top flanges of the two columns, forming the column B, as shown in Fig. 17, have a diameter of $15\frac{1}{2}$ inches, and are fitted with six $1\frac{3}{8}$-inch steel bolts on a $11\frac{1}{2}$-inch pitch circle, and the bottom flanges for the connection of the columns to the bedplate have also a diameter of $15\frac{1}{2}$ inches with six $1\frac{1}{2}$-inch bolts on a $12\frac{1}{2}$-inch pitch circle. The bolts are spaced so that they will straddle the fore-and-aft center line. Seven inches from the top of the columns are the connections for the $1\frac{1}{2}$-inch tie rods, which are connected to lugs 11 inches from the lower end of housing A.

$$F_B \times L_C = 607838 \text{ inch pounds},$$
$$M_B = 324668 \text{ inch pounds},$$
$$(F_B \times L_C) - M_B = 283170 \text{ inch pounds}.$$

The modulus of section at the lower end of combined column B is $117.2 = Z_1$.

$\frac{283170}{117.2} = 2416$ pounds per square inch of section as tensile stress at the bottom of column B. Z_1 for the combined section of column B at the top end $= 80.15$.

$\frac{324668}{80.15} = 4051$ pounds per square inch of section as tensile stress at the top of column B.

The bolts in each of the upper flanges B have to withstand the moment

$$\frac{324668}{2} = 162334 \text{ inch pounds} = M.$$

In this case:

$$s_1 = \frac{M}{\left(y_1 + \frac{2y_2^2}{y_1}\right) + \left(1 + \frac{2y_2}{y_1}\right) y_3}$$

$$s_1 = \frac{162334}{5.75 + \frac{2 \times (2.875)^2}{5.75} + \left(1 + \frac{2 \times 2.875}{5.75}\right) \times 4.56}$$

$s_1 = 9145$ pounds for one bolt.

$\frac{9145}{1.057} = 8652$ pounds tensile stress per square inch of section of bolt at bottom of thread.

For the bolts in each of the lower flanges B we have the moment 141585 inch pounds.

Now is

$$s_1 = \frac{141585}{6.25 + \frac{2 \times (3.125)^2}{6.25} + \left(1 + \frac{6.25}{6.25}\right) \times 4.56}$$

$s_1 = 7666$ pounds for one bolt.

$\frac{7666}{1.294} = 5924$ pounds tensile stress per square inch of section of bolt at bottom of thread.

The tie rods connected to column B are not taken in account, while in compression; due to their small diameter compared with their length, they would bend before they would produce a great compressive stress.

The area of the piston is, as stated before, 4417.9 square inches, the admission pressure 28 pounds absolute, assuming 4 pounds back pressure we have:

$$4417.9 \times 24 = 106030 \text{ pounds},$$
$$W \times \cos. 40^\circ = \underline{32938 \text{ pounds}},$$
$$73092 \text{ pounds} = \text{load on frames}$$

due to the steam pressure on the down stroke.

On account of the inclination of column B and housing A, this force is increased to $73092 \times \frac{113.2}{112.5} = 73544$ pounds.

From the location of column B and housing A under the cylinder (see Fig. 17), we can assume that each takes one-half of this load, or 36,772 pounds.

Each upper bolt of column B is under a tensile stress, due to this force, of $\frac{36772}{12 \times 1.057} = 2899$ pounds per square inch of section at bottom of thread, and each of the lower bolts under a tensile stress of $\frac{36772}{12 \times 1.294} = 2368$ pounds per square inch of section at bottom of thread.

In like manner each upper bolt of housing A is under a tensile stress of $\frac{36772}{10 \times 1.294} = 2842$ pounds per square inch of section of bottom of thread, and each lower bolt of $\frac{36772}{16 \times 1.057} = 2168$ pounds per square inch of section at bottom of thread.

No shearing stress is manifested on the bolts due to the connection of tie rods t_A, while the engine is in ahead motion and in the extreme rolling position towards N.

The total stress on the bolts of housing A is found to be, for the top flange:

15803 pounds tensile stress due to bending,
2842 pounds tensile stress due to steam pressure,

18645 pounds per square inch of section at bottom of thread.

For the bottom flange we obtain:

15404 pounds tensile stress due to bending,
2168 pounds tensile stress due to steam pressure,

17572 pounds per square inch of section at bottom of thread.

For the total stress on the bolts of column B we get, on the top end:

8652 pounds tensile stress due to bending,
2899 pounds tensile stress due to steam pressure,

11551 pounds tensile stress per square inch of section at bottom of thread.

And for the lower end:

5924 pounds tensile stress due to bending,
2368 pounds tensile stress due to steam pressure,

8292 pounds tensile stress per square inch of section at bottom of thread.

If the conditions are reversed to the extreme, that is, the engine in astern motion and the rolling towards N for astern motion, we find, from the former conclusion, that the stresses are the same on housing A due to S and N; but in opposite direction, on column B, the stresses, due to S and N, have the same effect but the stresses due to the tie rods t_B, from the mode of connection, produce a bending movement on the top end of column B and the bottom end of housing A. These tie rods in this condition assist the housing A again at the top end, and the force which produces the bending is as before, 24906 pounds.

The bending moment on each of the top ends of the two columns, forming column $B =$

$$\frac{24906}{2} \times 112.5 \times \left[\left(1 - \frac{7}{112.5}\right)^2 - \left(1 - \frac{7}{112.5}\right)^3\right] = 10698 \text{ inch pounds.}$$

For s_1 we obtain:

$$s_1 = \frac{M}{\left(y_1 + \frac{2y_2^2}{y_1}\right) + \left(1 + \frac{2y_2}{y_1}\right) y_3}$$

$$s_1 = \frac{10698}{5.75 + \frac{2 \times (2.875)^2}{5.75} + \left(1 + \frac{2 \times 2.875}{5.75}\right) \times 4.56}$$

$s_1 = 302$ pounds for one bolt,

$\frac{302}{1.057} = 286$ pounds tensile stress per square inch at bottom of thread.

The compressive stress produced on column B from the force 24906 pounds, according to the diagram of forces is:

$$24906 : 81 = x : 112.5$$
$$x = 34592 \text{ pounds}$$

$\frac{34592}{2 \times 18.85} = 918$ pounds compressive stress per square inch of section.

For shearing stress on each of the upper bolts of column B we have:

$$\frac{24906 \times 113.2}{12 \times 112.5} = 2089 \text{ pounds.}$$

$\frac{2089}{1.485} = 1407 \times \frac{5}{4} = 1759$ pounds tensile stress, due to shearing force, per square inch.

The total stress on each of the active bolts on top end of column B is now:

286 pounds tensile stress due to tie rods t_B,
2899 pounds tensile stress due to steam pressure,
8652 pounds tensile stress due to bending,
1759 pounds tensile stress due to shearing from t_B,

13596 pounds tensile stress per square inch of section at bottom of thread.

On the bolts at the lower end we have:
2368 pounds tensile stress due to steam pressure,
5924 pounds tensile stress due to bending,

8292 pounds tensile stress per square inch of section at bottom of thread.

The $2\frac{1}{4}$-inch bolt, which connects each of the tie rods t_B, has to withstand a shearing stress of:

$$24906 : 81 = x : 138.6$$
$$x = 42617$$

$\frac{42617}{2} = 21309$ pounds, which is equal to $\frac{21309}{3.976} \times \frac{5}{4} = 6700$ pounds tensile stress per square inch of section.

The amount of additional stress on the bolts at the lower end of housing A from tie rods t_B is:

$$\frac{24906}{2} \times 112.5 \times \left[\left(1 - \frac{11}{112.5}\right)^2 - \left(1 - \frac{11}{112.5}\right)^3\right] = 11390 \text{ inch pounds, and}$$

$s_1 = \frac{11390}{34 + 2.35 + 40.464} = 149$ pounds on one bolt $\frac{149}{1.057} = 140$ pounds tensile stress per square inch of section due to bending.

From the component of the stress on the tie rods t_B we obtain for the lower bolts of housing A

$$24906 : 81 = x : 112.5$$
$$x = 34592 \text{ pounds}$$

$\frac{34592}{16 \times 1.057} = 2045$ pounds tensile stress per square inch of section, and a shearing stress of 24906 pounds, equal to $\frac{24906}{16 \times 1.057} = 1473 \times \frac{5}{4} = 1841$ pounds tensile stress per square inch at bottom of thread.

The total stress on the bolts of housing A is now for the top flange:

15803 pounds tensile stress due to bending,
2842 pounds tensile stress due to steam pressure,

18645 pounds per square inch of section at bottom of thread, the same as before. The bolts in the bottom flange have to withstand a stress of:

15404 pounds tensile stress due to bending,
2168 pounds tensile stress due to steam pressure,
140 pounds tensile stress due to bending from t_B,
2045 pounds tensile stress due to component of t_B,
1841 pounds tensile stress due to shearing from t_B,

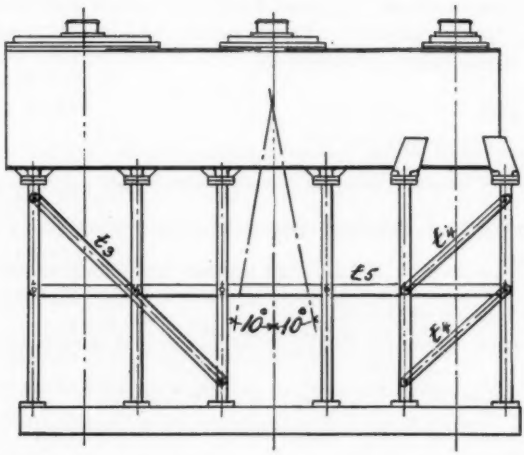
21598 pounds per square inch of section at bottom of thread.

Here is thus shown clearly the variation of stresses obtained by the different modes of tie-rod connection. Further it is shown that housing A is itself sufficiently strong; it is even strong enough if the tie rods were not considered, as the stress per

square inch of section would not be more than 7090 pounds per square inch due to the force $S + Z$.

The pitching of a ship can be classified into bow and stern pitching. The bow pitching, when the ship falls in the sea with the bow, has an effect upon the pitching tie rods of the engine as shown by Fig. 25.

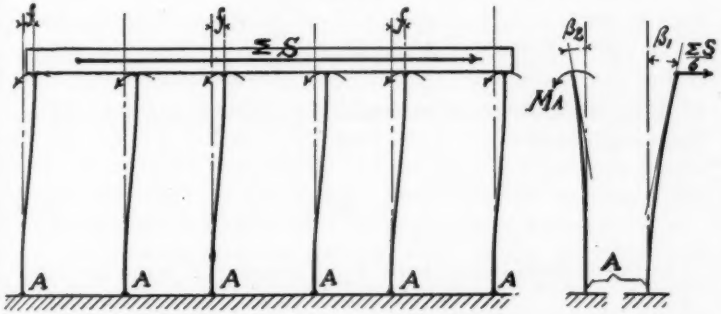
Fig. 25



The forward end of the engine being at the right of the figure, the ties t_4 will come in action, the tie rods being assumed to be used in tension only.

In Fig. 25 are shown the different arrangements of pitching tie rods; t_4, which are generally adopted for the back of an engine with forged frames, and are applied for bow and stern pitching. For the latter pitching, of course, they are placed opposite, t_5 is the strong-back for supporting the lower end of the guide faces. For the front of the engine the tie rod t_3 is commonly used for bow, as well as for stern, pitching. If no tie rods should have been provided, the action of the pitching would have an effect upon the frames as shown in Fig. 26.

Fig. 26



The angle β_1 of the tangent at the upper end of frame A (due to the force $\frac{\Sigma S}{6}$ with E_A as a constant modulus of elasticity), in which $S = \frac{W_1}{2}$ sin 10 degrees is the equivalent weight as before for one cylinder and Σ is the sum of the cylinders; $\frac{\Sigma S}{6}$ the force acting upon one frame A; and in this case, there being 6 frames A and 3 cylinders,

$$\beta_1 = \frac{\frac{\Sigma S}{6} L^2}{2 J_A E_A}$$

and the angle β_2 of the tangent (due to the bending moment M_A) at the same end,

$$\beta_2 = \frac{M_A L}{J_A E_A}$$

$$\beta_1 - \beta_2 = 0$$

$$\left(\frac{\Sigma S L}{6} - M_A\right) \frac{L}{J_A E_A} = 0$$

and

$$M_A = \frac{\frac{\Sigma S L}{6}}{2}$$

With an equal number of frames on each side we will have

$$M_A = M_B = \frac{\frac{\Sigma SL}{6}}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

If there should be on an engine 6 columns B and 3 housings A, we will have

$$M_A = \frac{\Sigma SL}{\frac{3}{2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16a)$$

The deflection of the columns B and housings A in the first case is:

$$f_B = \left(\frac{\frac{\Sigma S}{6} L^3}{2} - \frac{M_B L^2}{2} \right) \frac{1}{J_B E_B} = \frac{\frac{\Sigma S}{6} L^3}{12 J_B E_B} \quad . \quad . \quad . \quad (17)$$

$$f_A = \left(\frac{\frac{\Sigma S}{6} L^3}{3} - \frac{M_A L^2}{2} \right) \frac{1}{J_A E_A} = \frac{\frac{\Sigma S}{6} L^3}{12 J_A E_A} \quad . \quad . \quad . \quad (17a)$$

In the second case, for housing A, f_A will be:

$$f_{A1} = \frac{\frac{\Sigma S}{3} L^3}{12 J_A E_A} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

This f_{A1} will possibly not be equal to f_B, therefore the pitching tie rods must be of different sizes to suit the requirements for B and A respectively, to take up the pitching force ΣS and prevent the deflection due to pitching. Under certain circumstances housings A may be strong enough to withstand these stresses without tie rods.

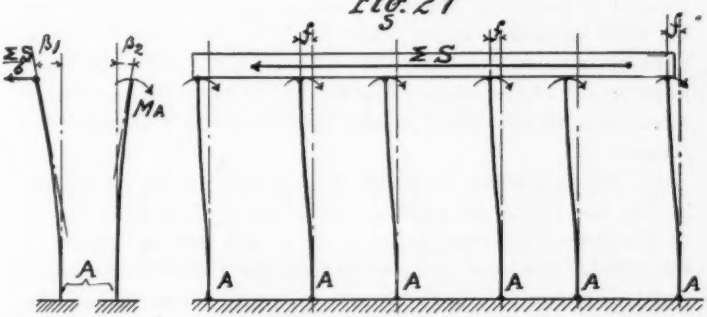
For stern pitching we can use the same equations, and tie rods t_3 will be in action (see Fig. 27).

For ramming action, the force ΣS_1 has to be considered.

S_1 is the equivalent of the moment of inertia of the weight of the cylinders at the ramming shock, and the tie rods t_4 have to be considered to take up the force due to this shock.

At the beginning of this article, in enumerating the stresses which act upon the frames, there were purposely omitted the stresses produced by the reverse shaft, because the reverse shaft should, if possible, not be carried in any way in bearings which are connected or secured to the frames. These bearings should be carried by the cylinder. The stresses produced by the valve gear and transmitted to the supports of the reverse shaft are of the kind which change from maximum + to maximum — with each revolution of the engine. Particularly in engines, which are built with slide valves for the low-pressure and intermediate-pressure cylinders, these stresses, under certain conditions, can be very severe, and are always acting in the athwartship plane, which is already the plane of greatest stresses upon the frames.

Fig. 27



Where saving of weight is a factor to deal with in the design of the engines, the reverse shaft is, however, in many cases carried on the frames of the forged-steel type. Under these conditions the columns which have to carry the bearings of the reverse shaft, should be increased in strength over their entire length in addition to the increase at the point of connection with the bearings. Not only must this be done, but the flanges at the top and bottom of the column should be made stronger, and the bolts of these flanges also, to withstand this continuously variable action of the valve-gear stresses, in addition to the stresses which are treated in this article.

From this investigation can be seen:

1st. That if the frames of a vertical inverted marine steam engine have for housing A and for column B an equal value of JE or $J_AE_A = J_BE_B$ and a crosshead guide connected to housing A only, the stresses produced by the rolling forces and by the crosshead force will not change their value for each frame under the condition of the engine changing from an extreme rolling position in go-ahead motion to an extreme rolling position in astern motion, but the direction of the forces will be opposite. The force Z is the reaction of a tensile stress in the cylinder bottom between the frames in both extreme positions.

2d. If the housing A and the column B have again the value $J_AE_A = J_BE_B$ and the frames are connected by a horizontal tie rod at the lower support of the guide force, while the upper support of the guide face is formed by lugs cast on the cylinder, the same stresses as stated before apply to both extreme rolling positions alike in housing A and column B, the direction being opposite in the two cases. The force Z is again the reaction or strain of a tensile stress in the cylinder bottom between the frames.

3d. If a case where the value of JE is different for the frames, so that housing A, on which is carried the guide face, according to the former assumption, has $J_AE_A > J_BE_B$ that of column B, the stresses produced as before remain the same for housing A and for column B in both extreme rolling positions, though the direction of these forces is opposite again in this last case, but the force Z, which is the reaction of a compressive stress in the extreme go-ahead position, changes to a reaction of a tensile stress in the extreme astern going position. By remembering this, the equation with the lower sign of $+$ and $-$ can be used for both extreme rolling positions to obtain the values of the stresses upon the frames; it is sufficient, however, to calculate the stresses in the extreme rolling position of the engine, being in ahead motion, to enable the designing of the frames.

4th. The connection of the tie rods as shown in Fig. 17 for tie rod t_A produces no extra stress on the frames, while the

connection of the tie rods t_1 and t_2, Fig. 3, increases the stresses on these frames.

I have not considered a rolling moment due to the inertia of W at the extreme rolling positions, because if rolling, the ship and with it the engines arrive from an upright position, with retarded motion, to a rest at extreme rolling position, and from this rest, with accelerated motion, to an upright position again, and *vice versa*. Sometimes a ship and with it the engines get a severe shock from the sea, even in upright position. This cannot be calculated, but the frames of the engines can withstand with safety these shocks, if they are designed to withstand the extreme rolling position.

ERRATA IN THE FIRST ARTICLE.

"Investigation of the Stresses on the Frames of Stationary Inverted Steam Engines," Volume XV, No. 1:

Page 23, line 4 from top, read $\left(\frac{l_2^2}{2} + l_2 l_3 - \frac{l^2 L}{2}\right)$.

Page 29, line 8 from top, read $\frac{\Delta_A}{a}\ \frac{\Delta_B}{a}$.

Page 30, in equation (11a) read $+\left(\frac{L_1 L^2}{2} - \frac{L^3}{6}\right)\sin 2\gamma\Big)\Big]$.

THE MECHANICAL LABORATORY OF THE ROYAL TECHNICAL HIGH SCHOOL OF BERLIN.*

By Walter Mentz,

Diploma Graduate of Charlottenburg Laboratory,
Associate, American Society of Naval Engineers.

In view of the great interest manifested in the United States at the present time regarding scientific investigation by German engineers, and especially their work in the Mechanical Laboratory of the Royal High School of Berlin, I take great pleasure, as an alumnus of that institution, and at the suggestion of the Council of the American Society of Naval Engineers, to give a short review of the arrangement and working of this laboratory.

As early as 1895 the State Department of Instruction determined upon the establishment of mechanical laboratories in the Prussian technical high schools, so that students, during their course of studies, could witness practical installations and workings of machinery, as well as learn how to conduct scientific tests.

At first the Minister granted about $11,000 for the building and $41,000 for the equipment, a total of $52,000 for the Mechanical Laboratory of the Royal Technical High School of Berlin. We may mention here that this is the proper name of the institution, although it is located at Charlottenburg, near Berlin, and only separated therefrom by the celebrated Tiergarten. To the English and Americans it is better known as the Charlottenburg Laboratory. A new professorship for the management of the establishment was created and tendered to Professor E. Josse, in October, 1896.

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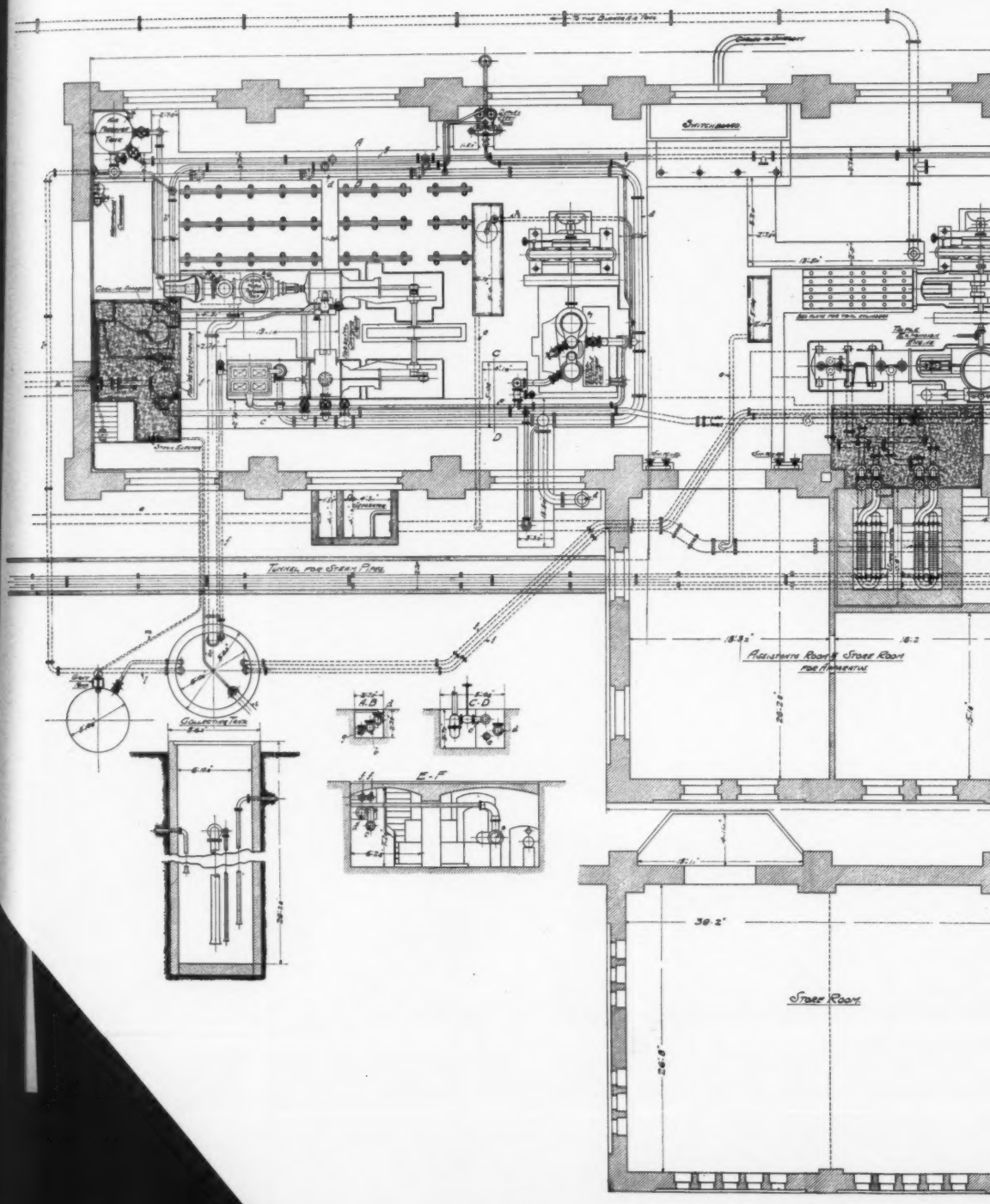
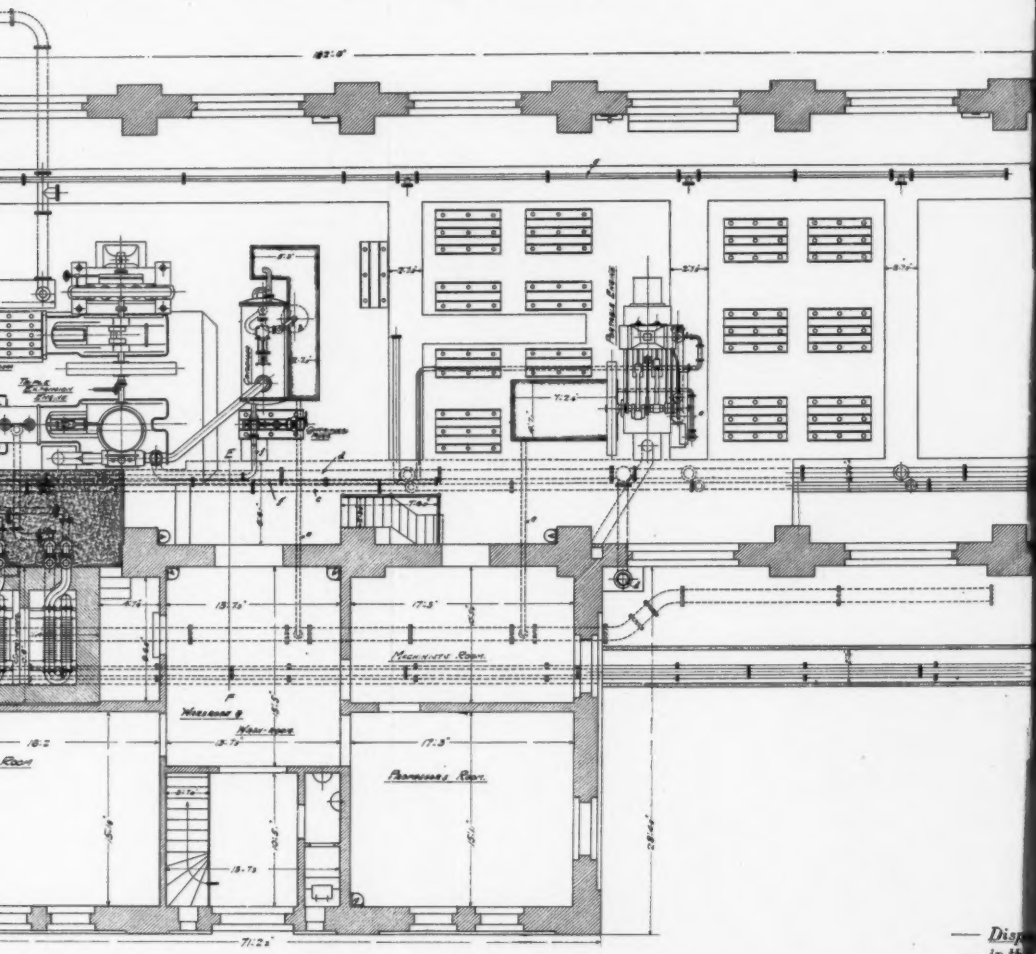
* Acknowledgment is made for the use of this data and illustrations to the Administrative Board of the Mechanical Laboratory, etc., of Berlin.


Switchboard
Tunnel for Steam Pipes
Assistants Room & Store Room for Attendants
Store Room
A-B
C-D
E-F
30-2"
26-8"

OTTENBURG MECHANICAL LABORATORY.



— Disp
— in th
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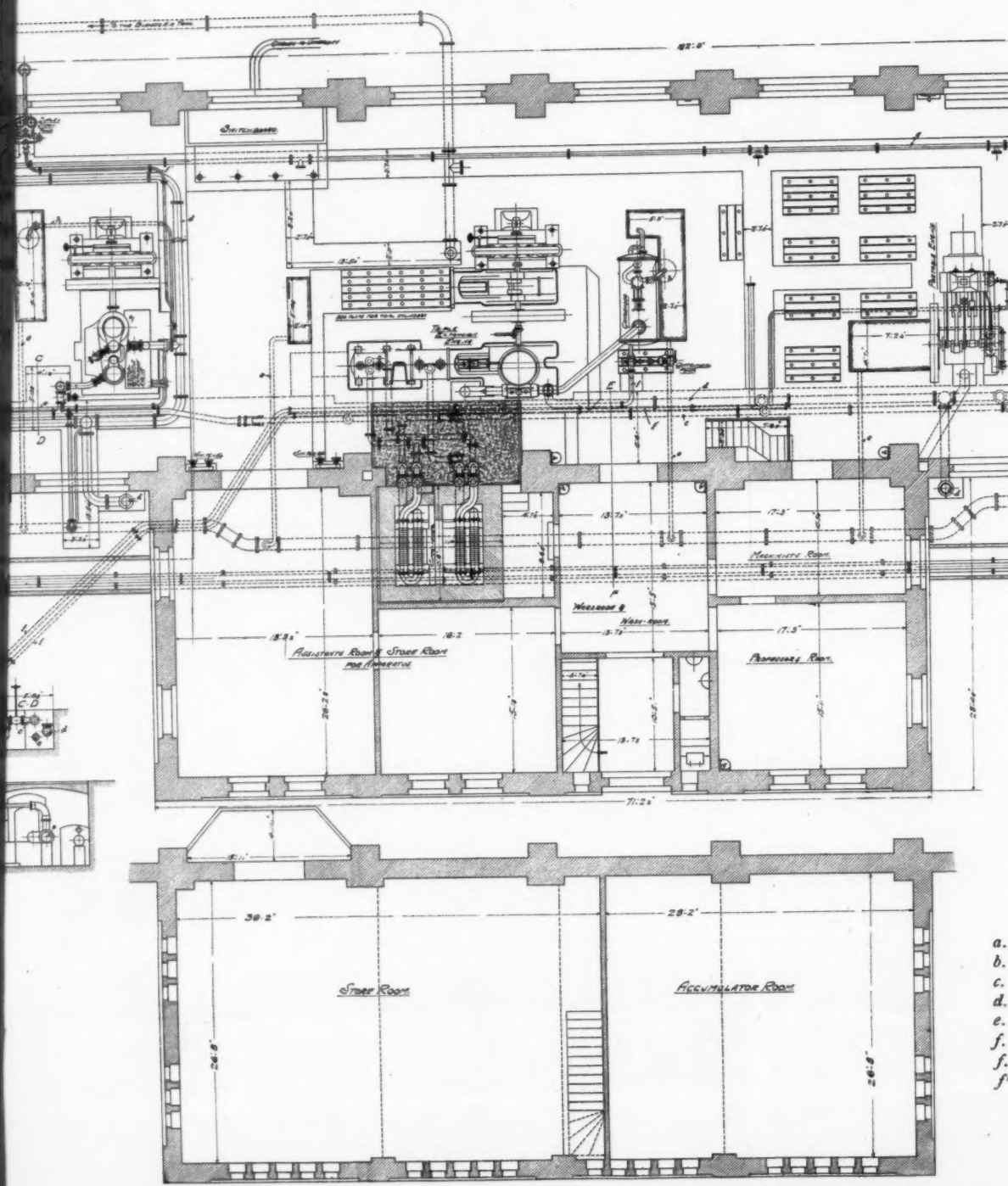
— of

a. Main steam line.
b. Live steam line (south).
c. Live steam line (north).
d. Exhaust line.
e. Waste water line.
f. Suction lines.
f. Injection water line for the portable
f'. Cooling water line for condensers of

CHARLOTTENBURG MECHANICAL LABORATORY.



a.
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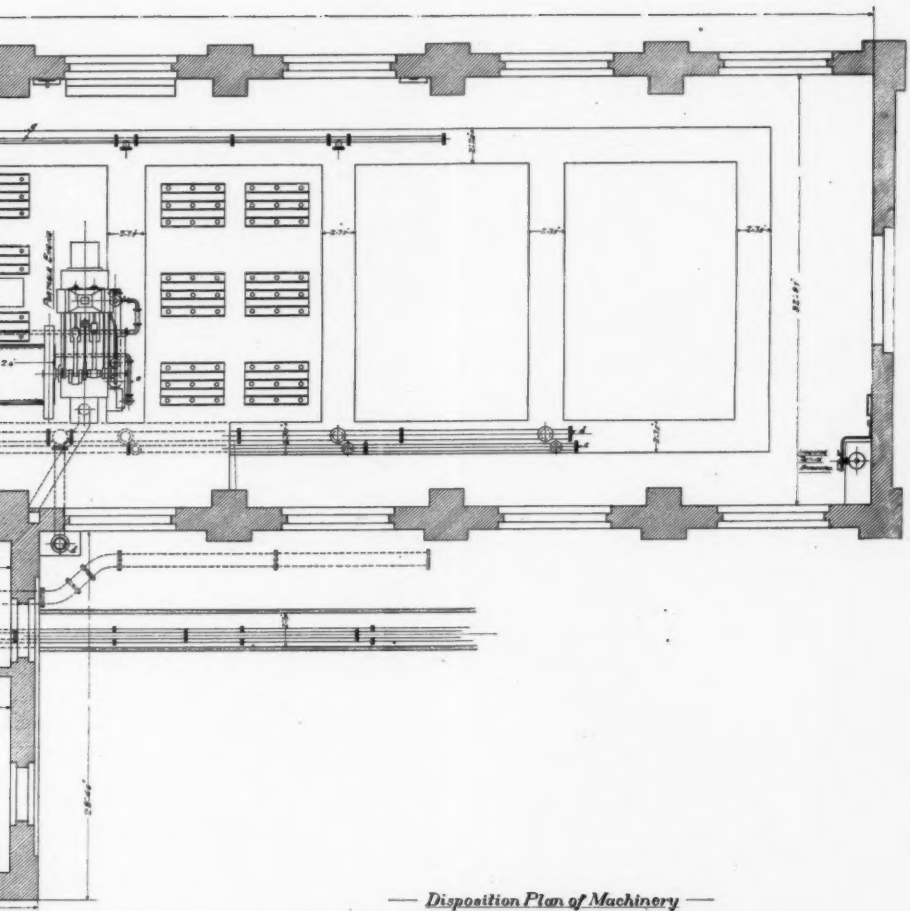


— Disposition Plan of Machinery —
— in the Engineering Laboratory —
— of the Royal Technical University —
— of Berlin (Charlottenburg). —

PIPE LINES.

a. *Main steam line.*
b. *Live steam line (south).*
c. *Live steam line (north).*
d. *Exhaust line.*
e. *Waste water line.*
f. *Suction lines.*
f. *Injection water line for the portable engine.*
f'. *Cooling water line for condensers of Engine One.*
g. *Pressure water and air lines.*
h. *Discharge lines from surface condensers.*
i. *Siphon lines.*
k. *Safety-pipes of the siphon lines.*
l. *Return lines of Engine Two.*
m. *Pressure air pipes to giant pump.*

GERMAN ADVANCE IN ENGINEERING RESEARCH.

At this time no German or foreign technical high school was in possession of any mechanical laboratory whose sole purpose was the scientific testing of machinery, and therefore the installation of such a laboratory had to be created along new lines.

The principal purpose of establishing such an institution was, of course, to answer the requirements of systematic instruction of students. Aside from this, it was kept in view from the beginning that an opportunity should also be extended to the laboratory faculty to make comprehensive scientific investigations, as well as conduct original research, and to afford facilities for the tests of any machinery that might be placed temporarily at the laboratory for such purposes.

Steam engines naturally took the first place as a subject for experimentation, not only because they form, so to speak, the foundation for instruction in machinery design, but also because the Berlin High School at that time possessed laboratories for the investigation of gas engines and electrical machinery.

The application of a load to these steam engines by brakes was dispensed with, although good power brakes, after the manner of inverted steam turbines, had been built, but such brakes were found very unwieldly and expensive in use. But since it would have been bad management to totally waste the energy of these engines, which were of considerable power, they were coupled as laboratory prime movers to the dynamos, pumps or air compressors.

The mechanical efficiency of these prime movers may perhaps be fairly established in this manner with quite a degree of accuracy, as the system offers the advantage that the absorbing machines themselves may become new subjects of useful tests.

GUIDING PRINCIPLES CONTROLLING ARRANGEMENT OF PLANT.

In the erection of the machinery the following guiding principles were observed. As the larger engines, in consequence of their greater original cost were likely to serve for a number of

years, they were built on solid foundations. As the smaller and cheaper engines might be displaced within a few years by improved types, they are, however, only bolted to iron ways which are built into the floors of the machinery laboratory so that each engine unit can be easily removed.

The first development of the laboratory was peculiar. According to the original plans the small means at command permitted only a hall of 60 feet by 33 feet, but in view of future enlargement, the height of this building was very great proportional to its area. This hoped for enlargement, however, was carried out much sooner than expected.

Professor Dr. Riedler, of the High School, presented the institution with several steam engines of a total value of $30,000, and the firm of A. Borsig, gave a steam boiler. After the Ministry of Instruction had made much needed enlargements of the laboratory building, it was proposed to use the large machines for the generation of electric lighting, the introduction of which, in place of the existing system of gas lighting, was much desired, and thereby presented an economical reason for urging a further increase of funds for maintenance and operation of the laboratory. The steam engines could then be kept in operation, charging accumulators during periods of instruction, and thereby reduce the cost of the generation of electricity to a minimum. The Ministry accepted this proposal, and gave $55,000 for this purpose, of which sum, of course, only a part accrued to the laboratory, as the expense of the electric-lighting installation had to be covered by this amount.

PLAN AND ARRANGEMENT OF THE LABORATORY.

The original length of the building was thus increased to 182 feet. At the present time the mechanical laboratory is being increased further, as it again has proven inadequate. The auxiliary rooms for measuring and testing apparatus, etc., were secured in an additional two-story building of 36 feet by 26 feet.

A plan of the building and the arrangement of the principal machinery is shown in the accompanying plate. Only such machines as are placed on solid foundations are shown; the smaller

machines are placed on the ways indicated, as may be needed. A number of trenches are built beneath the floor for the pipe circuits and cables. The pipe lines have branches at numerous places, so that a machine to be tested may readily receive or discharge, in the shortest way, steam, water or air, under pressure, or to exhaust.

In order to save the expense of a new boiler house and of a second chimney, the high-pressure boilers for the laboratory were placed into the boiler house that contained the steam-heating apparatus, and which was situated close to the laboratory.

There was a large boiler house which also had to be increased recently for the erection of a gas generator.

Of the boilers but three are used for laboratory purposes. There is a boiler of 861 square feet heating surface and 170 pounds steam pressure; a water-tube boiler with large tubes of 1,620 square feet heating surface for 255 pounds steam pressure, and a smaller water-tube boiler of 538 square feet heating surface and 142 pounds of steam pressure.

There is but one main steam line for all boilers. The steam pressure in the laboratory varies from 100 to 250 pounds, in accordance with the engines used and with the requirements of the tests that are being made.

The boilers are arranged with the necessary openings for the introduction of pyrometers. There are likewise facilities for taking samples of the products of combustion, for analyses of the gases, for the measurement of temperatures, etc. Several systems for the reduction of smoke have already been tried and are partly in use. A superheater for the generation of highly superheated steam is placed in the laboratory building.

The water required by the laboratory is taken in part from the city mains; but is principally obtained from a well outside of the building.

GENERAL DIMENSIONS OF THE PRINCIPAL EXPERIMENTAL STEAM ENGINE.

One of the largest and most valuable appliances of the laboratory, a part of the donation by Professor Riedler, is the quadruple-expansion engine built by the Vulcan Co., of Stettin. It is con-

structed very much after the marine type. The principal dimensions of this engine are:

Cylinder diameters, H.P., inches, ...	8.84
I.P. (1), inches, ...	13.15
I.P. (2), inches, ...	18.8
L.P., inches, ...	26.9
Stroke, inches	19.6

It develops 220 I.H.P. under 255 pounds steam pressure when making 150 revolutions per minute.

That the cylinders were placed tandem will be explained by the following:

The engine was designed with the special view of being used in tests to show the advantages of increased steam pressures, and therefore it had to be built as a quadruple-expansion engine. But as an arrangement of four cylinders side by side would have increased the frictional resistances too materially, besides occupying too much space for a comparatively small engine, two cylinders each were placed above one another. The lower cylinders are rigidly bolted together, but the upper high and intermediate cylinders are free, on account of the larger expansion from heat. The engine is fitted with "Klug" valve gear, frequently used in the engine designs by the Vulcan Co. It affords a better distribution of steam than a Stephenson link motion when the links are drawn in. One eccentric controls both valves of the two superimposed cylinders. On account of the double armed rocking levers interposed, the two upper cylinders have their steam chests between cylinders. All cylinders are steam jacketed. By a change cock placed on the exhaust pipe the engine can be made to exhaust into the condenser or into the atmosphere at will.

The automatic regulation of the engine called for by its being the driver of the electric generator, is by means of a governor with hydraulic-power cylinder especially designed for this purpose by Prof. Josse.

FACILITIES FOR DETERMINATION OF ENGINE STEAM CONSUMPTION.

This quadruple-expansion engine is, of course, especially adapted for tests of steam consumption and of heat measure-

ments. The manner of making such steam-consumption tests will be described more closely, as thus a fair idea will be had of the method by which instruction is carried on in the mechanical laboratory. The steam consumption is determined by measuring the quantity of water condensed by the surface condenser. This method is specially suited to laboratory practice, as it is possible thereby to obtain reliable results in a comparatively short time, provided the engine is working under a steady load. It is found that one hour is sufficient for making such a trial. The attachment and testing of indicators, the placing of scales, etc., takes some time, and as the indicator diagrams have to be measured with planimeter, and final calculations have to be made at the close of each trial, the students therefore require about three hours to make such a steam-consumption trial. This period cannot well be exceeded because the students have only a limited time at their disposal for such original work.

The engine is so arranged that it can be used as a compound, triple or quadruple-expansion engine. In the former case it runs with 55 to 85 pounds, in the second with 145 to 170 pounds, and in the last with 256 pounds steam pressure.

When the engine is used as a triple-expansion engine, the piston and the valve of the high-pressure cylinder are removed; the steam then merely passes through the high-pressure cylinder, which then operates as a steam separator. To work the engine as a compound, the piston and valves of the high and first-intermediate cylinders may be removed, or the valves only of these two cylinders are taken out and the steam is carried directly by a special pipe branch into the exhaust space of the first intermediate cylinder. But if it is desired to do away with the friction of the two pistons that are not doing work, then these pistons have also to be removed.

METHOD OF TEACHING STUDENTS HOW TO CONDUCT ENGINE TESTS.

That the students may acquire as much information as possible and become self reliant, they are placed in groups of five or six persons. Each group works by itself, under the direction of

an assistant, and each member of the group is thus fully occupied during the trial. Only one group is engaged on a special engine, although other groups may be engaged in studies and trials at the same time upon other engines or machines. Each group makes three steam-consumption trials on a special engine; first when it is working as a compound, then as a triple and last as a quadruple-expansion engine. The lubrication of the engine is attended to by an oiler, but otherwise the students have to do all the work, as, for instance, the attaching of indicators, the regulation of the engine to a given speed when the automatic governor is not in use, etc.

In order to obtain a steady working of the engine, the dynamo to which it is coupled during the trial is regulated with an adjustable constant resistance formed by incandescent lamps and arranged to absorb up to 200 I.H.P. The load on the dynamo may be changed by switches, in increments of 5 H.P.

During the first trial, the assistant directing the group explains the manner and method of the test and distributes the members of the group in such a way that, for instance, two students take diagrams on the two cylinders, one measures the condensed water, another observes the ampère meter, another the volt meter, while another measures the circulating water, etc.

At the second and third tests one student of the group is chosen as an assistant in charge, who has to station the other students and conducts the trials. Every five minutes diagrams are taken, and at the same time the data from the steam and vacuum gauges, revolution counter, ampère and volt meter and the temperature of the condensed and the cooling water, injection and discharge are taken and noted. For measuring the water of condensation two tanks are placed on scales, and between the two scales is a swiveled pipe, through which the air pump sends the water of condensation. Whenever one tank is filled to a certain weight of water, the pipe is changed about and the water is taken into the other tank. The condensed water from the several cylinders and valve chests is caught in four separate tanks. One student observes, every five minutes, the time in

which a tank of a given capacity is filled, and determines therefrom, the quantity of cooling water.

After about one hour the trial is interrupted, the diagrams are measured with planimeters, each student notes the mean effective pressure obtained on the several pistons, the value of the total water of condensation, the total cooling water, the mean values of ampères and volts, the revolutions and the temperatures of the condensed water and the injection and discharge water.

From the mean pressures each member of the group then determines by himself the indicated power. From the volts and ampères and the known degree of efficiency of the dynamo the effective work and degree of efficiency of the steam engine is obtained.

From the quantity of water condensed the steam consumption per I.H.P. per hour is computed. This value, at several trials in which the writer assisted, was 21.8 pounds, 17.75 pounds and 13.67 pounds, as the engine worked compound, triple or quadruple respectively.

From the data of the cooling water it is determined how many calories have been absorbed by it, and from the I.H.P. per hour is calculated how many calories have been transformed into work. If the test was carried out correctly, the sum of the calories used and carried off must be equal to those contained in the steam and the entrained water.

The water contained in the steam may be ascertained by a throttling calorimeter, the construction of which was described in "Zeitschrift Verein Deuts. Ing.," 1898, p. 664.

The engine may, of course, be worked as a non-condensing engine by disconnecting the air pump and exhausting into the atmosphere. The frictional resistances of the valves may be measured by means of the hydraulic-power cylinder provided in the design of the hydraulic governor.

Advanced students may make tests with saturated and superheated steam. The engine is therefore provided at all required places with special fittings for the reception of thermometers by which the temperatures of the steam may be measured on entering and leaving each cylinder.

DESCRIPTION OF THE TRIPLE-EXPANSION EXPERIMENTAL ENGINE.

The second largest engine of the laboratory is the triple-expansion engine, which develops 150 effective horsepower.

Whereas it is intended to work the quadruple-expansion engine principally by saturated steam of very high pressure, the triple-expansion engine is mainly built for superheated steam, it is therefore fitted with poppet valves and not with slide valves at the high and intermediate cylinders, as poppet valves are less affected by high steam temperatures.

The superheater is connected to the engine, and the steam for both the high and the intermediate cylinders is superheated. These two cylinders are arranged horizontally on the engine bed, and have a common piston rod; the low-pressure cylinder is arranged vertically to save space, and its connecting rod operates the same crank as those of the other two cylinders. Thus there are only two working connections, which must be regarded as an advantage to the triple-expansion engine of small power.

The principal dimensions of this engine are:

Cylinder diameters, H.P., 10.6 inches; I.P., 16.9 inches; L.P., 26.5 inches, and 19.6 inches stroke; revolutions, 150 per minute; steam pressure, 170 pounds.

In this engine also the high-pressure cylinder can be cut out so that the engine may work as a compound.

To enable the students to measure the clearance spaces easily and quickly, special valves are on hand which may be put in place of the admission valves. They are so constructed that water may be introduced to fill the clearance spaces of the cylinders and at the same time to provide for escape of air.

In addition to the main crank the shaft carries a second crank with an additional foundation plate, a crosshead and a connecting rod. On this foundation plate may be erected any desired power or working cylinder (pump, compressor, blowing or steam cylinder) and coupled to the engine by its special connecting rod. Generally, however, the load of this engine is transferred to an electric generator. When a pump cylinder or similar apparatus is connected, the generator runs empty without being excited, and its armature serves as a fly wheel.

The provisions for measuring temperatures, etc., are about the same as those of the quadruple-expansion engine.

The valve gear is arranged in such a manner that any desired steam distribution can be had and its influence upon the steam consumption can be determined.

For the determination of frictional resistances of the several moving parts the dynamo can be run as a motor, and the work consumed by the engine alone may be measured by electrical precision instruments more accurately than by indicator tests.

AUXILIARY EXPERIMENTAL ENGINES AND APPLIANCES.

Of other engines, steam pumps, etc., there are:

1. One horizontal compound engine of 60 horsepower with a pump attached. In this engine the high-pressure cylinder has a Rider cut-off, the low-pressure cylinder double-slide valves with Meyer's cut-off.

2. One compound condensing portable engine built by R. Wolf, in Madgeburg.

3. One vertical high-speed compound engine of about 40 horsepower. This engine, constructed by the Vulcan Co., of Stettin, is similar to a marine engine. It develops about 40 horsepower at 400 revolutions, and its main purpose is to give the students an opportunity to become skilled in taking indicator diagrams at higher speeds.

4. One single-cylinder engine with poppet valves, of about 40 horsepower.

5. One engine with double slide valves and Meyer's cut-off, of about 8 horsepower.

6 and 7. Two duplex feed pumps.

8. One simplex Blake pump.

The aim in the various constructions of all these engines has been to place before the students as many different types of valves and valve gears in actual working condition as possible. High or low-pressure cylinders may be used, each separately from the other, exhausting into the atmosphere or into a condenser at will. When the engine is running with one cylinder only and the valve-chest cover of the other cylinder is removed, students

may observe the running conditions of the various valve gears in the best possible manner.

The steam and exhaust-pipe connections are made by metal hose which can take pressures up to 284 pounds per square inch. These hose connections make it possible besides to suspend the entire engine from the traveling crane while in operation, and to demonstrate to the students the influence of different rates of speed upon the vibrations of the machine in the clearest manner possible.

EXPERIMENTAL APPLIANCES FOR HYDRAULIC TESTS.

In addition to the above smaller engines there are, of course, a large number of pumps, compressors, etc. Each of the two steam cylinders of the 60-horsepower compound engine under head 1 is coupled to a pump cylinder. These pump cylinders may be fitted with plunger pistons of larger or smaller diameter, so that the pump may deliver either 105 cubic feet of water at a pressure of 11 pounds per square inch, or 35 cubic feet of water at a pressure of 50 pounds per square inch.

The valve chest is provided with a number of openings covered by glass plates, and when a 60-candlepower incandescent lamp is introduced the students can observe the workings of the valves to great advantage.

The pump is provided with a set of valves of all well-known types. They can be placed in position readily, so that the students may observe not only the different constructions and workings, but also may take valve-lift diagrams.

The water discharged under pressure is taken up by a Pelton water motor, or by the water motor specially designed and presented to the institution by Professor Riedler, in which the water acts upon two single-acting plungers that transmit their motion to cranks placed at 90 degrees. But when these motors used as absorbers of work or pressure are not sufficient to obtain the required pressure, the water is discharged into a tank of special construction that contains a number of steel plates, each of which is provided with a small opening.

Besides this pump there is also an electrically-driven centri-

fugal pump which discharges about 200 gallons a minute at a pressure of suction and discharge equivalent to about 20 feet for each when making 350 revolutions per minute. The pump and electric motor can be readily uncoupled, and by braking the motor, its efficiency, and thereby the exact work done by the centrifugal pump, can be determined.

Borsig & Co., of Berlin, have placed at the disposal of the laboratory a compressed-air water lifter or trompe (so-called mammoth pump, described by Professor Josse in "Zeitschrift des Vereines Deut. Ing.," 1898, p. 981). It consists of a water pipe of 3.3 inches diameter which reaches deep into a well, and an air pipe 1.57 inches diameter near to it through which compressed air is forced into the water pipe. In the water pipe there is formed a mixture of air and water which is driven upward by the air pressure in the air pipe.

PNEUMATIC EXPERIMENTAL APPLIANCES.

Of pneumatic machines there are:

1. A blowing engine which can be coupled to the triple-expansion engine mentioned before, and which can discharge air at a pressure of 18 inches of water. By sliding the piston on its rod the distance between cylinder cover and piston can be increased from $\frac{1}{16}$ of an inch to $1\frac{1}{8}$ inches, and thus the effects of the clearance spaces can be readily studied.

2. A Westinghouse air compressor of the type used on locomotives.

3. A centrifugal blower electrically driven. The power consumption of this blower is also ascertained by first obtaining the efficiency of the motor and then braking the same.

The reason that so many electric motors have been installed in this mechanical laboratory, which is entirely separate from the electrical laboratory of the high school, is mainly because the mechanical engineer of today, even if he may have no comprehensive knowledge of electro technical subjects, must nevertheless be familiar with the principal conditions of electric motors. Then again because they are easily and quickly placed in connection with temporary installations of centrifugal pumps, etc.,

as prime movers, thus requiring merely a cable connection to the main line.

EXCEEDING IMPORTANCE OF A COMPLETE ELECTRIC INSTALLATION IN A MODERN ENGINEERING LABORATORY.

In order to simplify, for such students who have not acquired a special knowledge of electricity, the calculations of the work done by the electric motors, only direct-current motors are used. On these, electric motive force can be easily read from the pressure and quantity, while induction motors, though more suitable as motive power, are more difficult of computation. Besides, electric motors are specially adapted for application as dynamometers in testing centrifugal pumps, blowers, gearing, worm driving, etc.

That there is a storage battery has been mentioned before. It can completely absorb the work of the quadruple-expansion engine of 400 ampères at 250 volts and can return this output later for lighting purposes.

There have been added recently:

One single-acting steam air compressor, one Diesel motor and a suction gas motor which is combined with a SO_2 exhaust steam cylinder.

The mechanical laboratory has already tested a number of outside or trial machines. The Riedler-Express pump and the special valves used by Riedler and Stumpf in their blowing engines were first tried for quite a while in the laboratory before they were given over to the public. Prof. Josse has made extensive tests on the utilization of the heat contained in exhaust steam with the so-called exhaust-steam engines. (See "Zeits. Ver. Deutscher Ing.," 1901, p. 1077.)

To the 150-horsepower triple-expansion engine was added a cylinder with valve motion, this cylinder using as its working medium sulphur dioxide (SO_2). The surface condenser of the steam engine then acts as a boiler for the SO_2 operating in this cylinder. This exhaust steam-heat utilizing device and cylinder has developed 32.2 per cent. of the work of the steam engine

and thus reduced the consumption of superheated steam from 11 pounds per I.H.P. per hour to 8.245 pounds.

ADMINISTRATIVE AND TECHNICAL STAFF OF THE LABORATORY.

Besides the director, Prof. Josse, there are, one superintending engineer and, on an average, about seven assistant engineers employed in the laboratory. The boilers are managed by one chief fireman with four assistants; the engines are operated by one chief machinist, four machinists, one electrician, seven mechanics and laborers.

The method of instruction is plainly seen from the somewhat detailed description of the steam-consumption trials.

To pass the graduating examination, which is the final goal of the Prussian high-school student, a study of at least eight semesters is required. Students in mechanical, naval and marine engineering must have had one year's practice in the workshops of these respective branches before entrance.

The students generally take part in the exercises of the laboratory during three semesters in periods of one afternoon each week for each group. These exercises are preferably taken in the third, fifth and eighth semesters. Such tests as are general or which have been made by all graduates are given in a table at the end of this article.

Advanced students or engineers who desire to make special investigations of machines can take part in a special course provided for such purposes.

THE LABORATORY AN EDUCATIONAL NAVAL ENGINEERING FACTOR OF IMPORTANCE.

One purpose of the mechanical laboratory of Berlin, at Charlottenburg, consists in the fact that all of the younger naval and marine engineers are required to take some practical part during the course of their studies in scientific tests of the most varying character. Therefore, besides having a clear insight into the management of tests and a better understanding of the subjects discussed at the lectures, they are able themselves to make the scientific tests and trials which are now so often

required on naval and merchant vessels, and by all larger private shipbuilding establishments of Germany, and to conduct these independently and in a perfectly reliable manner.

This mechanical laboratory is, therefore, of especial importance and value. It has long been recognized that in most fields of mechanical engineering, enduring progress can only be attained by scientific tests and theoretical deductions therefrom, and not by one-sided theories or from mathematical formulations.

The educational work that it has done for the German Navy and mercantile marine is very great, even though done in an indirect manner.

EXERCISES IN THE MECHANICAL LABORATORY FOR STUDENTS OF THE THIRD SEMESTER.

A. *Investigations of elements of machinery.*

Tests of flange joints and packings. Calibration of steam and vacuum gauges and indicator springs. Investigations of worm and belt drives, gearing and dynamometers.

B. *Investigations of motors and smaller machines.*

Taking apart and reassembling apparatus.

Determination of the efficiency of electric motors and smaller steam engines by brakes of various types.

Determination of the efficiency of smaller pumps, centrifugal pumps, blowers, Pelton wheel, etc.

Management of revolution counters, tachoscopes, tachometers, anemometers and other measuring instruments.

Manipulation of indicators, stroke reducers of various types, planimeters and electrical measuring instruments.

Determination of the indicated work of electrical machines, pumps and simple steam engines.

Determination of the efficiency of the machines tested.

EXERCISES IN THE MECHANICAL LABORATORY FOR STUDENTS OF THE FIFTH AND SIXTH SEMESTERS.

A. *Steam Boilers.*

Management of boilers; water-tube and fire-tube types.

Study of the processes of combustion; analyses of gases; value of fuels; draft and temperature measurements.

Feed apparatus; manipulation of the same. Steam pumps, injectors.

Systems of smokeless firing and firing by generator gas.

Evaporation tests; steam-pipe lines, losses by condensation and reduction of pressure in them. Arrangements to compensate for expansion. Steam traps, etc.

B. *Steam Engines.*

Removing and reassembling the several parts.

Investigations of various valves and valve gears.

D-valve, Trick and piston valves; Meyer and Rider gear, poppet valve gear, Collmann's old and new gears, Klug link motion.

Investigations of condensers and air pumps; jet and surface condensers.

Management of steam engines. Working one, two, three and four-cylinder engines.

Steam-consumption trials on single and multiple-cylinder engines, non-condensing and condensing, with saturated and superheated steam of various pressures to 256 pounds pressure.

Study of the heat processes in steam engines. Heat balance. Influence of steam jackets and reheaters. Determination of moisture in steam with the throttling calorimeter. Regulation of steam engines. Velocity ratios for load variations.

C. *Water Pumping Machinery.*

Pumps and their management. Investigation of a pumping engine. Determination of its capacity; the volumetric and mechanical efficiency at various speeds.

Changes in heights of suction and pressure. Investigation of various valves, automatic and controlled; valve movements; valve-lift diagrams.

The same investigations for high-pressure pumps. Air chambers.

Syphon pipe lines with ejectors for the removal of air.

Mammoth pump. Centrifugal pump.

D. *Hydraulic Engines.*

Hydraulic motor with variable admission and reversing, etc.

Pelton motor.

E. *Blowing Engines.*

Ventilating fans. Compound air compressor with and without water injection; with or without jacket cooling; effect of inter cooler.

Blowing engines with various types of valves and valve gears.

Westinghouse air compressor.

F. *Air-pressure Engines.*

Working with cold air, with or without expansion, with preheating by products of combustion and steam injection.

G. *Refrigerating Machines.* (In preparation.)

Production of refrigeration, management of refrigerating machines, measurements of their capacity, tests.

H. *Transmission of Power.*

a. By steam.

b. By water pressure.

c. By air pressure.

d. By electric current.

Determinations of efficiencies and economies of power transmissions under varying circumstances.

COALING AT SEA.

THE CUNNINGHAM-SEATON PROPOSED METHOD.

By Civil Engineer A. C. Cunningham, U. S. N.

One of the most important problems of modern naval warfare is the question of coaling ships at sea. The subject is engaging the attention of all powerful maritime nations. Extended experiments have been conducted to determine the rapidity and reliability of such appliances, the trolley system of conveyance devised by Mr. Spencer Miller having probably attained the greatest degree of success.

The most interesting proposition recently made is that upon the part of Great Britain to build four colliers having a speed of 18 knots, of a carrying capacity of 10,000 tons of coal each, and equipped with the latest coaling devices. Such colliers would become part of and accompany a fleet whose radius of action would thereby become greatly extended and whose movement could not be definitely predicted by an opposing force.

Numerous and widely scattered coaling stations cannot entirely solve the fuel problem in time of war, for it may not always be possible to secure coal from such fuel depots. Even though the coaling station might only be a short distance from the scene of operation, there would be many occasions when it would be hazardous for a Commander to weaken his blockading or cruising fleet by sending some of the vessels to coal.

The coaling station must be strongly defended, and as coal is a contraband article, the fuel depots could not be kept supplied by neutral powers during time of war.

Were it necessary for a ship to begin expending her ammunition from the time of putting to sea until her return to port without constant and certain means of replenishment, there would be no rest until the defect was remedied; and yet this weakness, however, is the existing condition as regards the fuel

supply upon which the war ship relies to reach or evade an enemy, and the defect is none the less serious on account of our familiarity with repeated and vexatious delays in coaling.

To appreciate the fuel question one has but to review the history of our West Indian campaign during the war with Spain. From beginning to end the most important and leading subject was that of fuel supply. One of our first measures was to secure every ship available for carrying coal. An absorbing question was whether the *Oregon* could secure coal to bring her to the Cuban coast in time for anticipated battle, or whether she would have coal enough to escape a possible pursuit from a superior force. The maintenance of the blockade of Santiago depended entirely upon whether or not our ships could be kept supplied with sufficient fuel of fair quality.

Previous to the Spanish-American war we had practically no devices for coaling at sea, nor had the problem received much serious study on the part of our naval administrators. Coaling with a float between the ships to keep them apart was found to be dangerously impracticable with any motion on the vessels whatever, and we were obliged to establish a base in the enemy's country where smooth water was assured.

Even bales of cotton between two ships will be forced into their plating in a seaway, and the only expedient that has been reported as at all successful is a large mass of green brushwood held in a compact mass between the ships and acting as a spring fender.

Although coaling over the side is the most rapid and con venient method, it has been amply demonstrated that it can not be done in a seaway with any rigid body interposed between the ships to keep them from contact.

As it is impossible for two ships to lie safely alongside of each other in a seaway with any rigid connection between them, the method of coaling at sea so far experimented with has been with one ship towing the other and using the trolley system devised by Mr. Miller. The battleship *Illinois* has recently been fitted with the latest improved development of this appliance.

With the trolley system the battleship tows the collier at a

distance of about 1,000 feet. A line passes from the mast head of the battleship over the mast heads of the collier, and is kept taut by a sea anchor. Conveyors carrying bags are passed back and forth over this line by conveying lines operated by winches. Compensating devices are used for maintaining uniform tension and to allow for the varying distances of the ships. It is reported that in foreign experiments with this device that thirty-five to forty tons of coal have been conveyed per hour in a

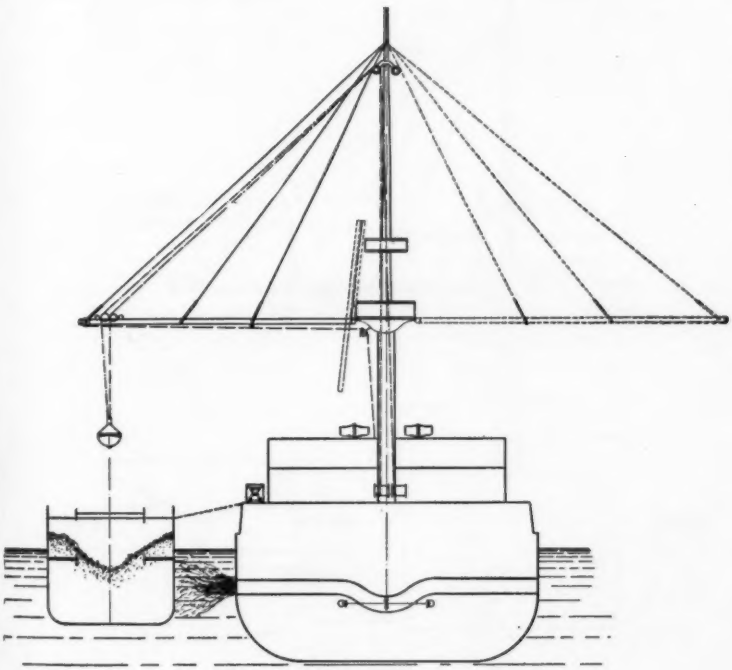


FIG. 1.

COALING AT SEA. CUNNINGHAM-SEATON PROPOSED METHOD.

Showing entire apparatus installed on war ship. Dotted lines indicate coaling boom rigged in, or transferred to opposite side. Reversible water-jet apparatus taking water from one side and discharging on the other.

moderate sea and even in half a gale of wind and with a speed of towing of from 8 to 11 knots an hour.

The author has designed an apparatus for use in coaling ships at sea by means of which vessels may safely lie alongside each other for the purpose. The general principles of the apparatus are illustrated in Figs. 1 and 2.

In Figs. 1 and 2 the jet apparatus is shown as a separate in-

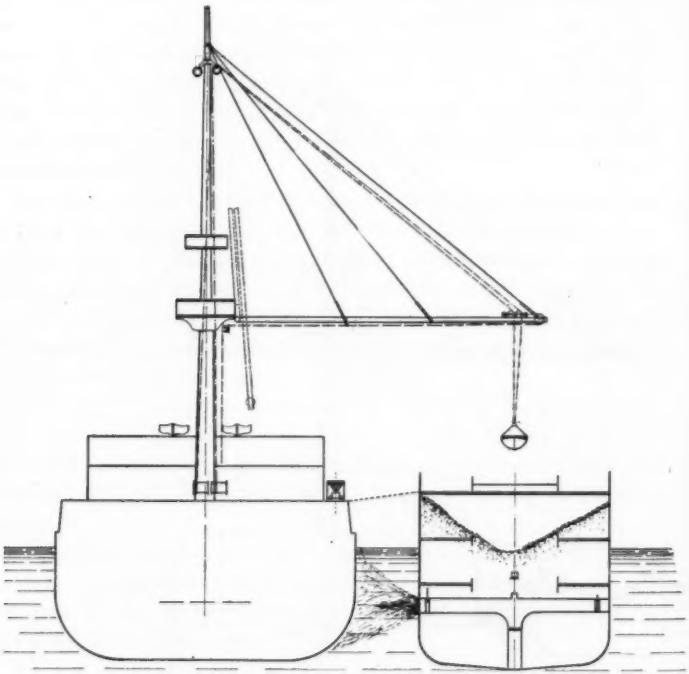


FIG. 2.

COALING AT SEA. CUNNINGHAM-SEATON PROPOSED METHOD.

Showing conveyor apparatus installed on war ships and water-jet device on collier. Dotted lines indicate coaling boom rigged in. Water-jet device installed to take water through bottom and discharge at either side.

stallation. The circulating, fire and bilge pumps on a ship may, however, be utilized to produce the jets.

The method of coaling in tow must prevail until it is demonstrated that a more satisfactory method can be devised for coaling along side. Coaling from a vessel in tow is, however, funda-

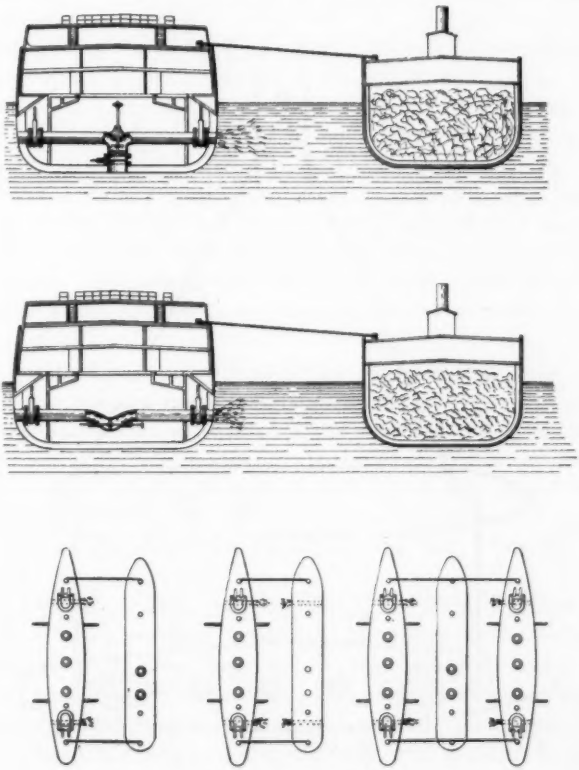


FIG. 3.

COALING AT SEA. CUNNINGHAM-SEATON PROPOSED METHOD.

Showing general location and different installations of water jets, and various combinations of war ships and colliers using same. The availability of water jets for maneuvering purposes is apparent in plan.

mentally objectionable, as it requires much sea room, can not be operated on a collier without masts, and the parting of a tow line will wreck the whole system.

With the apparatus projected the ships are kept from touching each other and are maintained at the required distance apart by

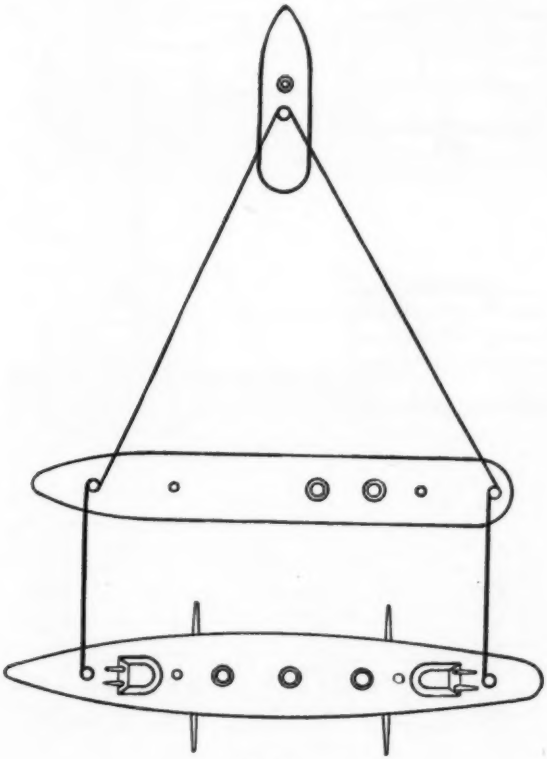


FIG. 4.

COALING AT SEA. CUNNINGHAM-SEATON PROPOSED METHOD.

Extemporized application of the water-jet principle using small vessel to produce lateral motion of the system. Particularly applicable with tugs and coal barges.

means of water jets which are the equivalent of elastic struts. The required distance between the vessels is secured by lines passed between the ships, for there are no rigid connections of any kind. It is not intended with this apparatus that any appreciable lateral speed should be produced; a tendency to separate the vessels sufficient to keep the connecting lines taut is all that is desired or necessary.

The general location of the water jets and various combinations of ships fitted with the same are shown in Fig. 3.

Neither is it necessary that the jet of water should reach from one ship to the other; the reaction of the jet upon the surrounding water is sufficient to produce the tendency to separate. The momentum of the water jets is imparted to the ships by reaction, and when a tendency to move or a slight motion has been produced the momentum of the ship for resisting wave action is very great. The resistance of the ship to which the connection is made to moving sideways through the water completes the combination.

Theoretically, if time were of no consideration, very small jets would produce the necessary momentum.

An extemporized application of the water-jet principle, using a tug to produce the lateral motion, is shown in Fig. 4. In emergency a collier may be temporarily fitted with the apparatus by installing centrifugal pumps on deck forward and aft and leading the suction and discharge pipes over the side when required for use.

Mr. William Seaton, M. E., of the C. W. Hunt Co., has proposed a coal-conveying apparatus for use with the water-jet device, which is also illustrated in Figs. 1 and 2. The lines having been passed between the ships and the water jets turned on and regulated, the boom is rigged out and coaling proceeded with. For the most rapid operation the coal is taken from the hold of the collier with a grab bucket, dumped into a hopper on the ship, and from thence conveyed to the coal chutes in cars.

The advantages of this apparatus are in being connected to one ship only and in having independent positive motion, thus permitting it to work without regard to the rolling or pitching of

the ships, or of varying distances between them, as the vertical and lateral motion of the bucket are always under control. The bucket and hopper are not essential to the apparatus and the coal can as readily be conveyed in bags, only more slowly. When bags are used, the conveying apparatus can as well be on the collier as on the ship to be coaled, or a conveying apparatus can be rigged upon each mast of both the collier and war ship.

By giving to the water-jet apparatus as much power as installed in an ordinary tug boat, valuable maneuvering properties may be secured which have not heretofore been possessed by ships, and thus make the water jets of value at all times. The tactical diameter would be reduced by such an installation. Direction of the vessel could be changed or maintained in action or on blockade without longitudinal motion. Docks could be approached and left with greater facility and safety, and a purely lateral motion could be obtained when desired.

Although any one who purposes to place additional weight or occupy additional space in our already crowded war ships may expect a cool reception, there are many considerations which indicate that such an installation as here projected should be made on the war ships. It would then become possible to take coal from any kind of a collier without the latter having special fittings. Coal could also be taken from a captured vessel, or a war ship could transfer her coal to another vessel of the fleet in case of emergency.

The simplicity and compactness of these proposed devices, the ease with which they can be installed on existing ships, and the general value of the water jets for maneuvering purposes, should commend the proposed method to naval authorities.

CHARACTERISTICS OF THE NORMAND WATER-TUBE BOILER.

By Augustin Normand, Jr., Associate.

The Normand boiler is a development of the design invented in 1876 by Commandant Du Temple (Fig. 1), the distinctive feature of which is an assured circulation obtained through large return or downcast tubes which are not exposed to the flame. In other respects, however, there is but little resemblance between the original Du Temple design and the early form constructed by M. Normand.

Monsieur Normand continued improving the Du-Temple boiler by modifying many details of construction, and a more efficient design was adopted in the French torpedo boats of the *Flibustier* class, under the name of the Du Temple-Normand boiler (Fig. 2). In this early type of steam generator the motion of the hot gases was in a vertical direction, the smokestack being placed about the middle of the length of the boiler.

About 1890 M. Normand practically gave the marine engineering world a new type of marine steam generator. He enlarged the size of the tubes, modified the form of the steam drum and improved the joint of the tubes in the drum. In place of having the flame follow a vertical direction he conducted the gases horizontally between the rows of tubes before permitting them to enter the stack. Marked efficiency was secured as a result of these changes. This improved boiler was fitted in some Japanese and French torpedo boats. Since then there has been progressive improvement in the details of construction, and thus in 1894 there was built the famous French torpedo boat *Forban*, a craft which attained a speed of 31.03 knots—a speed that had never been before realized by either naval or merchant craft.

In the boilers of the *Forban* the smokestack is placed at the

front end of the furnace. The flame is arrested at the back of the grate by a bridge wall forming a complete screen. The gases return towards the front by distributing themselves among the rows of tubes throughout their whole length before reaching the funnel. This is known as the direct-flame boiler, and this type

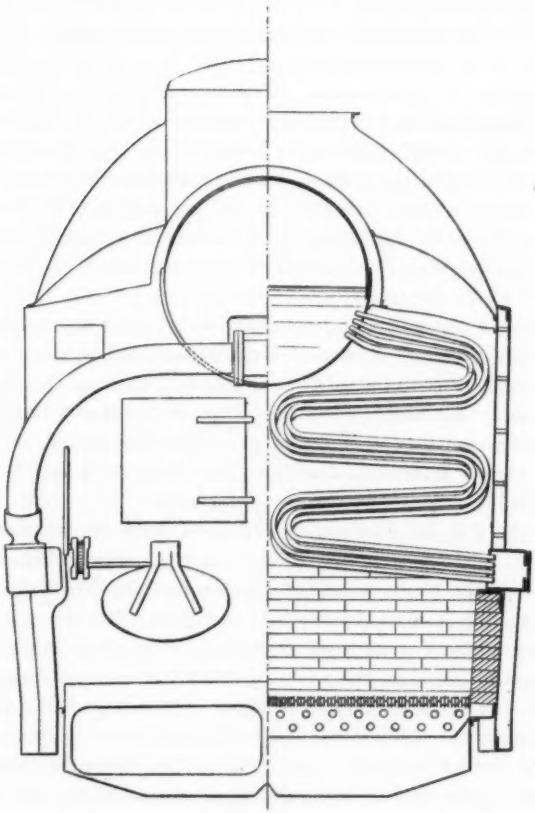


Fig. 1

DU TEMPLE BOILER.

of steam generator is both economical and efficient. It likewise possesses endurance (Figs. 3, 4, 5).

It is the actual relative performance of boilers which determines their comparative worth, and therefore the results of some trials made by us in 1896 are given, which will show the efficiency obtained even at that time. These trials were made under closed-stokehold conditions on the boilers of the Russian torpedo boat *Pernoff*, and continued for two hours, the grate surface having been reduced 0.37 in area during the second trial.

	1st trial.	2d trial.
Coal burnt per square meter of grate, kilos	318	196
foot of grate, pounds	65	40
Water evaporated per kilogram or per pound of coal	6.97	11.6
Temperature of feed water, centimeters	28	28
Fahrenheit	82	82

The following is the performance of the boilers of the celebrated cruiser *Bogatyr*, a vessel of 6,500 tons displacement, constructed by the Vulcan Company, of Stettin, who have secured the right to manufacture boilers of this design in Germany. These results show the adaptability of the boiler for installation in large cruisers. These trials occurred on June 20, 1902, and therefore show the efficiency of the Normand boilers of the latest design.

PERFORMANCES OF THE CRUISER *BOGATYR*.

No. of the trial.	Speed, knots.	I.H.P.
1	23.34	20,365
2	23.72	20,530
3	23.22	19,725
4	23.52	20,023
Mean	23.452	20,160.75

The first trial was one of six hours' duration, and took place in smooth water between Dantzig and the Island of Bornkolm. The primary purpose of this trial was to determine the coal consumption at maximum speed.

The second trial was likewise of six hours' duration, and there

was nothing done to the machinery between the runs except to clean the grate.

Very careful boiler data, however, were taken during trials that took place on succeeding days, so that the efficiency of the boilers could be ascertained. The following are the principal data obtained during those trials:

	Date of trial.	
	June 22d.	June 23d.
Pressure of air in the fire room, millimeters	14	15.5
inches	.56	.62
Horsepower of main engines	20,196	20,343
Number of revolutions per minute	151.6	150.85
Speed in knots during trial	24.9	24.21
Consumption of coal per hour per I.H.P., kg	0.646	0.655
I.H.P., pounds	1.42	1.44

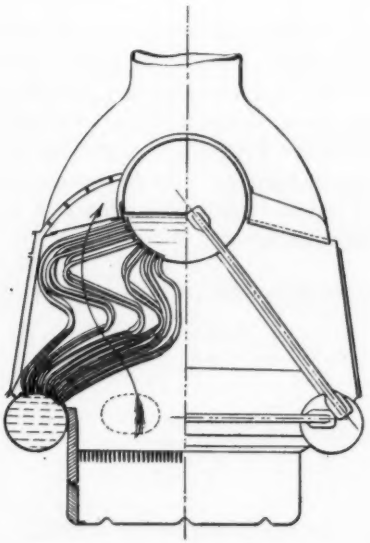


Fig. 2

DU TEMPLE-NORMAND BOILER.

There was at all times a reserve of steam, showing that the capacity of the boilers had not been reached. The contract requirements were largely exceeded, as will be noted by the following summary:

	Contract requirements.	Official performances.
Speed	23	24.15
Pressure of air in fire room, millimeters	52	15
inches	2.08	.60
Consumption of coal per hour per I.H.P., kg	0.91	0.65
I.H.P., pounds	2	1.43

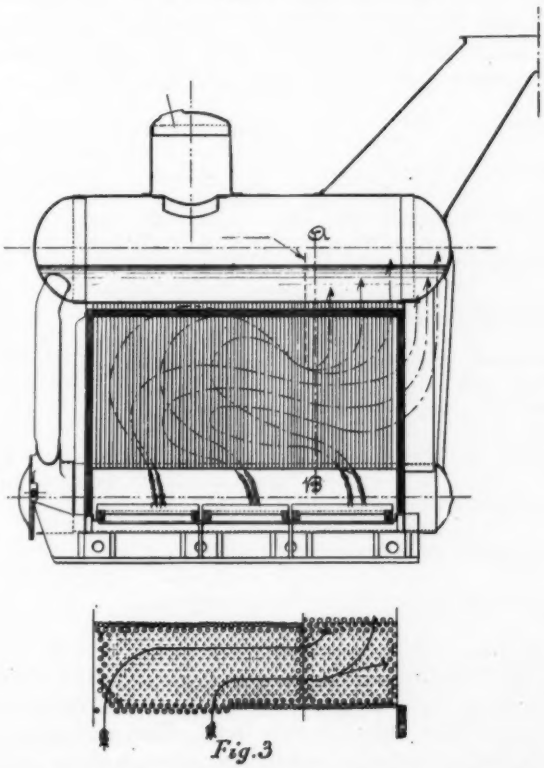


Fig.3

NORMAND BOILER. DIRECT-FLAME DESIGN. LONGITUDINAL SECTION.

Such official results are the best tributes that can be paid to the efficiency and capacity of boilers of this design.

The latest and highest development of the boilers of M. Normand place the smokestack forward of the elements instead of in the rear. There is also a modification in the disposition of the bridge wall, so that this screen causes the path of the gases to return to the front (Fig. 6, Fig. 7, Fig. 8).

Careful tests were made with the torpedo boat *Lancier*, of 128 tons displacement and 1,400 horsepower, and likewise with the torpedo boat *Tourmente*, of 132 tons displacement and 1,500 horsepower, to determine the efficiency of the Normand boilers installed within them. Both these boats were built in 1879.

	First trial. T. B. *Lancier*.	Second trial. T. B. *Tourmente*.
Grate surface, millimeter^2	3.162	3.627
square feet	34.04	38.72
Heating surface, millimeter	149.70	154.65
square feet	1,610.77	1,664.03
Coal consumed during the trial, kilos	1,968	2,657
pounds	4,329.6	5,846
Coal consumed per hour, kilos	954	1,328.63
pounds	2,098.8	2,923
Coal consumed per square meter of grate surface per hour, kilos	300	365
Coal consumed per square foot of grate surface, pounds	61.3	74.63
Water vaporized per hour, liters	7,750	9,950
pounds	17,061	22,904
Water vaporized per kilo of coal, kilos	8.14	7.5
pound of coal, pounds	8.14	7.5
Water evaporated per square meter of heating surface, kilos	51.77	64.3
Water evaporated per square foot of heating surface, pounds	10.9	14.18

The following results were obtained with a boiler of 1.81 square meters of grate surface (19.48 square feet), and 80 square meters of heating surface (860.8 square feet). The working pressure was 10 kilos (142 pounds), and the temperature of the feed water was 25 centimeters (77 degrees Fahrenheit).

Trial.	Duration in hours.	Rate of combustion.		Vaporization.	
		Kilos per square meter of grate.	Pounds per square foot of grate.	Kilos of water per kilo of coal.	Pounds of water per pound of coal.
1st	12	98	20	10.94	10.94
2d	4	199	40.8	8.214	8.214
3d	2	300	61.4	7.694	7.694
4th	7	97	19.8	10.03	10.03
5th	6	200	40.9	8.035	8.035
6th	6	300	61.4	7.654	7.654

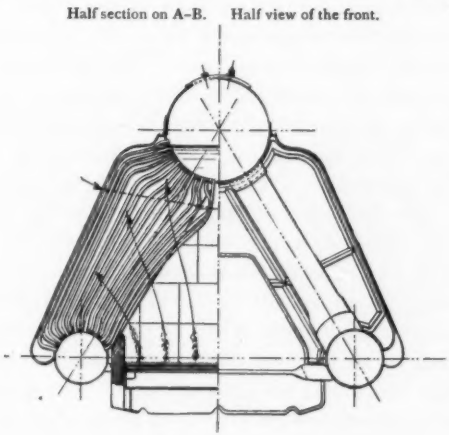


Fig. 4.

DIRECT-FLAME DESIGN.

ADVANTAGES OF THE NORMAND BOILER.

1st. It is strong and is able to withstand great variations of conditions. It works as reliably and satisfactorily with coal consumptions of 300 to 400 kilos of coal per square meter of grate as when there is only a consumption of 100 kilos per square meter of grate. The remarkable results secured on the large cruisers *Châteaurenault*, *Le Montcalm*, etc., where a combustion of 200 kilos per square meter of grate has been secured

under normal conditions, show the manner in which the boilers can be forced. Very few boilers of any design present such elasticity of construction.

2d. This design of boiler has relatively large tubes, the interior diameters being respectively 25, 30 and 34 millimeters (.98, 1.18 and 1.34 inches). The body of these tubes is alone exposed to the direct action of the flame. The tubes are fitted to the drums without any joints except simple expanding, which insures a firm hold and perfect steam tightness. A special tool permits withdrawal of the tubes without damaging them, so that the same tubes after withdrawal can be replaced in the boiler. Another tool has been designed whereby the tubes can be tested individually under a high hydraulic pressure. The ability of the tube to withstand excessive pressures is thus guaranteed.

3d. The design of the boiler permits a very rapid and reliable circulation, and this circulation is of a nature that keeps the interior of the tubes clean. The circulation is so regular and so positive that the formation of steam pockets or the overheating of the metal is absolutely prevented. The efficiency, capacity and endurance of the boiler is thus maintained. In an article read before the British Society of Naval Architects in 1895, Mr. Normand showed the manner in which steam pockets could be prevented, and in accordance with the principles contained in that paper his boiler has been constructed. The guiding principles in designing water-tube boilers as enunciated by Mr. Normand are as follows:

(a) The direction of the flow of water in the tubes, especially in their lower and more heated parts, must be as vertical as possible.

(b) The circulation of the water must be active.

(c) The ratio of length to diameter of the portions subjected to excessive heat must not be too large.

(d) The sectional area of the outside downtake of water-return tubes from the upper to the lower reservoirs ought to be of large proportions, in order to permit an increase in the intensity of circulation. Extended laboratory experiments have demonstrated that the coefficient of transmission of heat in tubes varies

from 1 to 5, according to whether or not the circulation of water within the tubes is very rapid.

4th. The space required for its installation is comparatively small, since this design of boiler is used on torpedo boats and the smallest type of craft.

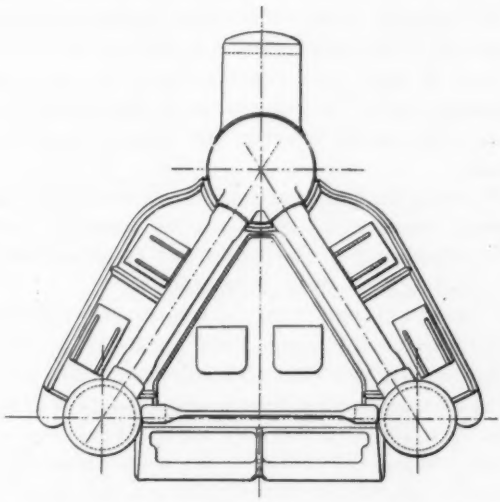


Fig. 5.

DIRECT-FLAME DESIGN. FRONT VIEW.

5th. Its weight is comparatively small, for with a grate surface of 7.07 square meters (76 square feet) and a weight of 19.2 French tons (18.9 English tons), without water, and 22.7 French tons (22.24 English tons) with the boilers filled, Mr. Normand succeeded in producing 3,600 I.H.P. Such a design of boiler made it possible to install in the latest type of French torpedo-boat destroyers, two boilers whose evaporative capacity was sufficient to produce 7,200 horsepower. Two boilers of this type installed in the French torpedo boat *Arquebuse* of 300 tons displacement and 6,000 horsepower made it possible for the

vessel to obtain a speed of 30.75 knots, while in the *Arbalete*, a torpedo-boat destroyer of the same dimensions, a speed of 31.37 knots was secured. The value of such lightness in boiler construction is a distinct military advantage, providing endurance is not sacrificed thereby. It should be ever remembered that weight saved in boiler construction of battleships or armored cruisers permits the designer either to increase the speed of the vessel, extend her radius of action, or install heavier armor or armament.

6th. There is ample and complete elasticity, since the tubes are sufficiently curved to yield to the considerable variation of temperature that exists between the various portions of the tubular nest.

7th. The extended experience that the Normand firm possesses in designing, constructing and operating tubulous boilers has shown the expediency of the means and facilities that are provided for cleaning such steam generators.

When performing regular service torpedo boats generally get under way with comparatively light pressures in the boilers. Steam rises rapidly, however, in such boilers, and in a very short time after getting up anchor the maximum speed may be obtained. It is often the case that the torpedo boat must come to anchor very unexpectedly, without any precautions having been taken to reduce the thickness of the fire. Fires have been suddenly hauled after the anchor has been dropped, and in less than a half hour the engine secured and the pressure removed from the boiler without subjecting the latter to injury or strain.

Should any accident happen to the engine compelling the boat suddenly to stop, it is only necessary to open the furnace and combustion doors and to close the ash-pit doors to check evaporation. Such is the character of the strain to which these generators can be *repeatedly* subjected.

8th. From the standpoint of economy there is no water-tube boiler that is its superior, as shown by the results of official trials of boilers of this design.

Messrs. Normand and Sigaudy have designed a battery installation of boilers modeled on the Normand type coupled back-to-back. The furnaces are placed end-to-end as in the case of the

double-ended Scotch design. Such a design of boiler gives a large grate surface proportional to weight and space occupied. It reduces the number of boiler attachments as well as the amount of piping required. Where such an installation is desired for vessels of large size, the interior diameter of the tubes should be between 30 and 34 millimeters (.98 and 1.18 inches). Since the boilers of such vessels must have endurance, it is essential to give the tubes a thickness of 3 or 4 millimeters (.12 to .16 inches).

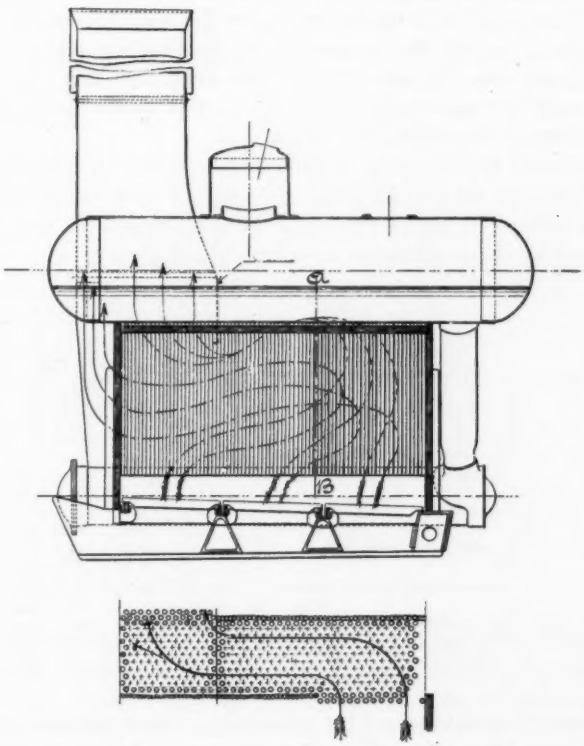


Fig. 6.

RETURN-FLAME DESIGN. LONGITUDINAL SECTION.

In providing a Normand-Siguady design of boiler for vessels of large tonnage it is essential to arrange for a high rate of combustion, since such ships at times are expected to make high speed. A guarantee that 180 kilos of coal per square meter of grate per hour will be consumed is given (36.1 pounds per square foot of grate). The construction of this form of boiler, however, is of such character that in case of necessity the combustion can be forced to the same degree as in the case of the express boilers of torpedo-boat destroyers.

As compared with the Scotch boiler the weight is much less, even when basing the comparison on burning 100 kilos of coal per square meter of grate (20.5 pounds per square foot of grate), and such a consumption is attainable even under natural-draft conditions in the Normand design.

As to the actual results secured from boilers of the Normand-Sigaudy type, the following data secured by the Societé Anonyme des Forges et Chantiers de la Méditerranée will be read with interest by those interested in the boiler problem.

Naval service.	Name of vessel.	Nature of trial.	Duration, in hours.	I.H.P.	Consumption of coal.			
					Square feet of grate.	Square meter of grate.	Kilo per hour per H.P.	Pounds per hour per H.P.
France.....	*Dunois*.....	Power.	3	7,709	55.2	270	.842	1.85
		Coal.	18	2,710	16.55	81	.725	1.6
	Lahire.....	Power.	3	7,222	44	240	.816	1.8
		Coal.	18	2,668	14.6	71	.639	1.4
	Châteaurenault...	Power.	3	23,229	28.88	142	.747	1.64
		Coal.	24	14,934	16.55	81	.660	1.45
	Montcalm.....	Power.	3	19,798	33.33	163	.815	1.79
		Coal.	24	11,320	16.55	81	.672	1.48
Portugal...	*Sao Gabriel*.....	Power.	2	3,710	38	186	.903	1.98
	Sao Rafael.....	Coal.	6	1,144	8.18	40	.633	1.39
Spain.....	*Rio de la Plata*...	Power.	2	6,420	35.37	173	.876	1.92

Some of these vessels have made long cruises since the above trials, and have returned to home ports with boilers in highly efficient condition.

The *Rio de la Plata* after her acceptance trial made a fifteen

months' cruise in the waters of the South Atlantic and the Pacific oceans. During that cruise the vessel steamed 24,000 miles and burned over 3,500 tons of coal of various sizes and qualities under the most variable conditions as regards sea and weather. During that period the boilers were under natural and forced draft of various pressures, and at no time was a tube obstructed or disabled, or did any mishap occur to a boiler appliance.

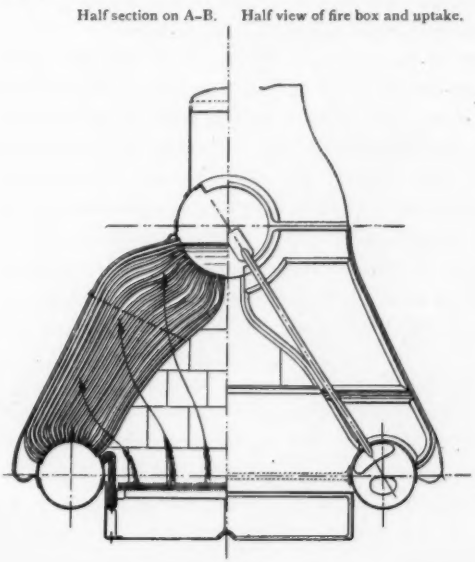


Fig. 7.

RETURN-FLAME DESIGN.

In August, 1901, there was published in the "Revista General de Marina" an article by Lieutenant Manuel Carballa, in which the author highly commended the efficiency and endurance of the Normand-Sigaudy boilers installed on board the vessel of which he was the executive officer.

The war in South Africa influenced the Portuguese government to send to the vicinity of Lorenzo-Marquez, the cruisers *Sao*

Gabriel and *Sao Rafael.* These vessels cruised in that neighborhood for a considerable period, and upon the return to Lisbon an examination of the boilers showed that they were in excellent condition. The endurance of these boilers was conclusively shown, since they were not in need of overhauling.

The *Montcalm*, on which the President of the French Republic made a voyage to Russia in the spring of 1902, steamed through the Baltic with only one-third of her fires lighted. As her commanding officer was somewhat apprehensive that the vessel would not reach Cronstadt at the hour that the *Montcalm* was expected by the Russian officials, orders were given to force the fires, and it was but a short time before the other vessels of the squadron were left in the rear. During an hour and a half of this forced-draft run, in the effort to reach Cronstadt on time, there was burned in the furnaces, without the least inconvenience, 225 kilos of coal per square meter of surface (46 pounds per square foot). It can be well realized that boilers capable of being forced to this extent without subjecting them to injury are military weapons of great value, and in time of war can be depended upon to render splendid service to the Navy and the Nation.

When the Normand small-tube boilers, which have proven so successful in torpedo boats and destroyers, were proposed for installation in cruisers and battleships, some doubts were expressed in regard to the advisability of installing such boilers in such vessels. The work performed by small-tube boilers in the *Sao Gabriel*, *Sao Rafael*, *Montcalm* and other vessels designed for extended cruising undoubtedly shows that they are suited for installation in cruisers and battleships. Not only has their endurance, efficiency and capacity of evaporation been proven, but it has been shown that the design admits facility in stoking that will greatly lighten the labor of those working in the fire rooms.

The installation of these boilers in vessels of the mercantile marine has been successfully accomplished, although their use in ships of that character is somewhat limited. The performance of the tug *Adour*, of the port of Bayonne, an installation that was effected as far back as 1895; the work of the tugs *St. Mammes* and *Corbeil*, which ply on the upper Seine, and which have jet

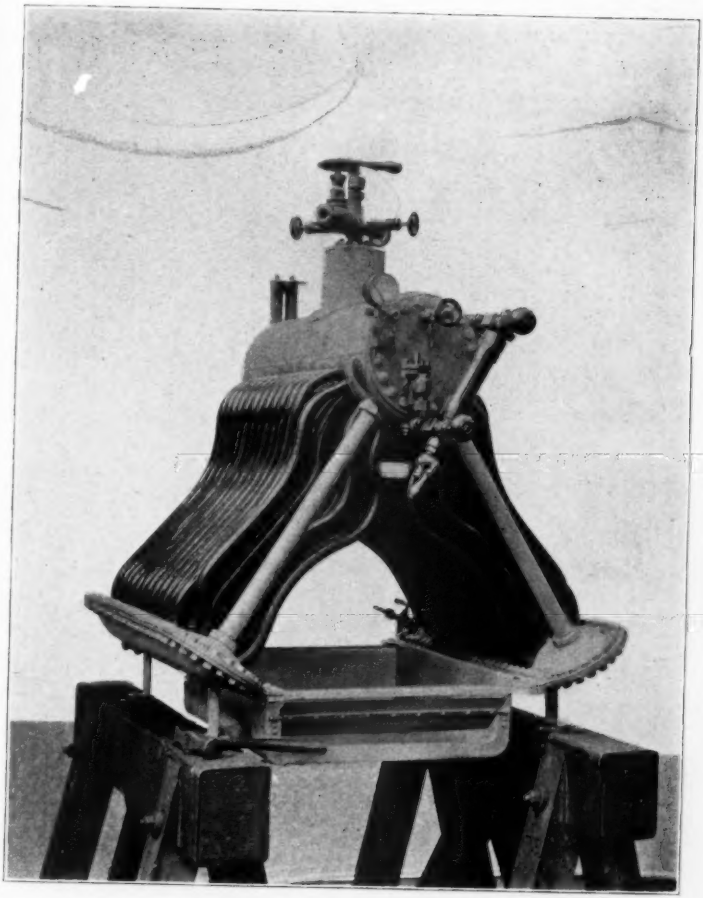


FIG. 9.

BOILER FOR YACHT "HELENE."

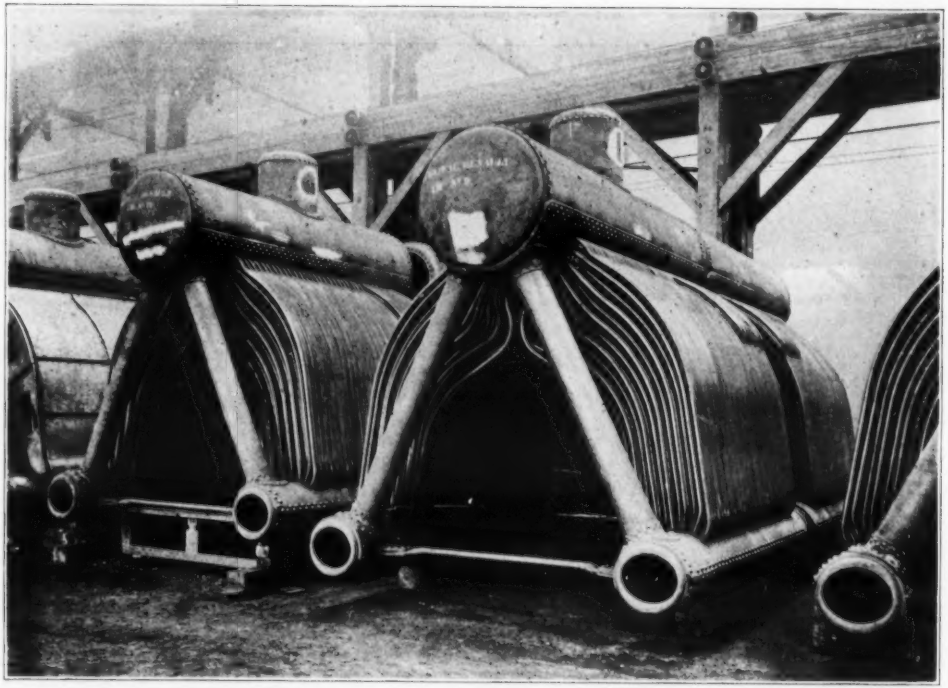


FIG. 10.

CHANDIËRE NORMAND-SIGAUDY BOILERS.

condensers; the yacht *Helene* (Fig. 9), and the passenger boat *La Touques*, of Havre, as well as its application in several steam launches, show the adaptability of the boiler for installation in traffic and passenger craft.

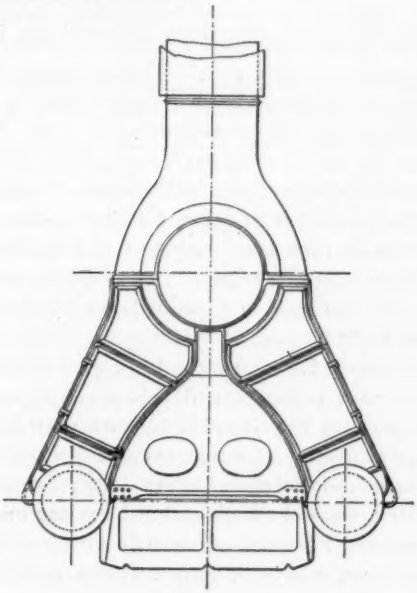


Fig. 8.

RETURN-FLAME DESIGN. FRONT VIEW.

Mr. Sigaudy, in an article read before the British Society of Naval Architects, during the Jubilee year of Queen Victoria, pointed out the adaptability of such boilers for installation in the largest of the transatlantic liners. The suitability of the boiler for installation in cargo steamers can be conclusively shown.

It may be further mentioned that the installation of these boilers for manufacturing and industrial purposes has been effected in the fitting shops of the Societé des Forges et Chantiers at Havre, and for the operation of the pumps of the Pontaniou docks at Brest. (Fig. 10.)

DIRECTION OF ROTATION OF PROPELLERS.

By Nino Pecoraro, Engineer, First-Class, R. I. N.

Translated from "Revista Marittima", August-September, 1903, by Assistant Engineer R. E. Carney, U. S. Navy, Retired, Member.

For some years many of the battleships and large cruisers of the British Navy have been equipped with propellers arranged in such manner as to turn from outboard to inboard when going ahead; in other words, the upper tips of blades move towards the hull, so that the starboard propeller is left-handed and the port propeller right-handed.

It is a well-known fact that hitherto propellers have generally been made to turn outward, and this practice, established by custom and perhaps not justified by any particular reason of convenience, was followed for many years. In some cases, however—and particularly for going astern—the propellers have been made to rotate toward the hull, and, if I am not mistaken, those of the *Palestro* and *Varese* of our own Navy rotate in that direction. These were, however, only isolated cases, and it was, perhaps, due to the acknowledged harmful influence of the system upon the maneuvering qualities of the ship that it was speedily abandoned.

The question has now presented itself under a new aspect. When the propellers are arranged to turn towards the hull it is very easy in the case of modern ships to install the engines in such manner as to locate the working platform near the middle of the vessel, that is, near the central longitudinal bulkhead which separates the two engine rooms. Such an arrangement is of advantage, because it renders it easier to operate and watch the engines and to interchange orders between the two engine rooms, and also insures greater convenience of installation and better conditions of ventilation for the working platform. And these

advantages have led to a renewed investigation of the question of the direction of rotation of propellers.

Aside from the slight detriment to the maneuvering power, which is not considered of sufficient importance to cause one to relinquish the advantages inherent in the better installation of the engines, another question has arisen which raises new doubts and uncertainties. What influence may the change of direction of rotation of the propellers exert upon the efficiency of propulsion? Obviously, the detection of such a harmful influence as loss of efficiency should have greater weight than any other consideration. In order to ascertain whether such influence exists, Mr. R. E. Froude has made numerous experiments with models in the famous Haslar tank, and the results of these experiments were set forth by him in an interesting paper read before the Institution of Naval Architects in 1898.

Froude's experiments were made with models of three battleships, twelve cruisers, five destroyers, and one vessel of very shallow draught and with an overhanging stern of the Thornycroft pattern. From these experiments Froude arrived at the conclusion that by making the propellers turn inboard a slight gain in efficiency was generally attained. From the means of the results of the different experiments made it was concluded that the mean gain in efficiency achieved by causing the propellers to turn inboard instead of outboard was about 1.5 per cent.

It may not be amiss, before proceeding further, to call attention to some details of the experiments and to some variants in computing the efficiency which Froude has deemed it necessary to use in view of the particular accuracy required in the investigations made, it being a question of estimating differences of very small amounts.

It is known that in Froude's original method of propulsion experiments with models, which method is set forth in the "Transactions of the Institution of Naval Architects," 1883, Vol. XXIV, the computation of the efficiency of propulsion (or the ratio $\frac{f.c.e.}{f.c.a.}$ in case of the model) is made on the basis of the following considerations:

Let n, s, t, be respectively the number of revolutions, the thrust and the torsional force necessary to obtain with a given type of propeller the propulsion of the model at a speed v. Under these circumstances the propellers work in the wake or train of friction which follows the ship at a given speed. These propellers, therefore, will not have, as regards the water surrounding them, a speed v, but a speed $v^1 < v$, and theoretically they will work in the same conditions in which they would work at a speed v^1 in water undisturbed by the passage of the hull. Hence, the number of revolutions being equal, the thrust and the torsional force t will be the same for propellers working behind the hull towed at a speed v as for propellers working by themselves at a speed v^1. In the first case, however, the travel per revolution will be $\frac{v}{n}$, and in the latter case $\frac{v^1}{n} < \frac{v}{n}$. The two rates of travel will therefore be in the ratio of v to v^1, and hence the turning power will be equivalent to torsional force $\times \frac{2\pi R}{\text{travel}}$, in which $R =$ radius of pulley upon which the force t is exerted.

In the former case the turning power f will be less than the turning power f^1 in the latter case. That is to say that the performance* of the propellers working in conjunction with the hull at a speed v will be greater than that of the propellers working by themselves at a speed v^1 in the ratio of f^1 to f, or in the ratio of v to v^1.

The final expression of the total efficiency of propulsion is $\frac{r}{f}$, if r is the resistance to towing of the hull model by itself at a speed v. This efficiency may also be put in the form $E = \frac{r}{s}\frac{s}{f}$, wherein $\frac{r}{s}$

* The expression of the performance or of the efficiency of the propellers is

$$e = \frac{s \cdot v}{t \cdot 2\pi R \cdot n} = \frac{s \cdot \frac{v}{n}}{t \cdot 2\pi R} = \frac{s}{t \cdot \frac{2\pi R}{\frac{v}{n}}} = \frac{s}{f}, \text{ hence } f = t \cdot \frac{2\pi R}{\frac{v}{n}},$$

which is exactly what is called "power of rotation."

represents the increasing resistance due to the action of the propeller, and $\frac{s}{f}$ the efficiency of the propeller itself working behind the hull at a speed v. According to the preceding reasoning, however, this efficiency is equivalent to the efficiency e of the propeller by itself at a speed v^1, and at a number of revolutions n multiplied by $\frac{v^1}{v}$. We have, therefore

$$E = \frac{r}{s} \times \frac{v}{v^1} \times e.$$

Hence, to sum up, the value of the total efficiency of propulsion at a speed v is given by the product of $\frac{r}{s}$ (increase of resistance due to the action of the propeller), of $\frac{v}{v^1}$ (coefficient of wake) and of e (efficiency of the propellers by themselves corresponding to the number of revolutions n at which the model propulsion takes place). The total efficiency, according to Froude's original method, is determined by estimating by experiments, with hull and propeller models in conjunction with each other, the value of the "increase of resistance" corresponding to the propulsion, then by computing, by some suitable method, the value of the wake coefficient $\frac{v}{v^1}$ and multiplying these two coefficients (the product of which constitutes what is termed hull efficiency) by the performance of the propellers at a number of revolutions n, which performance is to be determined separately by means of experiments with propeller models by themselves.

In what has been said about the effect of the wake current upon the action of the propellers a hypothesis is admitted which deserves special attention. It has been assumed that propellers working in conjunction with the hull at a speed v give, at a number of revolutions n, the same thrust which they would give if they were working by themselves at the same number of revolutions and at a speed v^1, and it has further been assumed that

in order to attain this thrust the same torsional force is required in the two cases.

This hypothesis would be correct if the action of the wake consisted merely in giving to the water a forward speed uniform throughout the whole region in which the propellers operate, or, in other words, if the wake were composed of parallel particles of water endowed with the same speed. But these conditions are not strictly fulfilled. And in fact, if the hypothesis referred to were correct, the curves of thrust and torsional force determined with the propellers working in conjunction with the hull at a speed v would practically coincide with the same lines determined with the propellers by themselves at a speed v^1. But a difference, though practically of slight importance, may always be noted between the curves in question, and while bearing in mind that such difference is partly due to the method of computation by which the value of v^1 is determined, it cannot be denied that it must be attributed mainly to the assumption of uniformity of wake. Another confirmation of the imperfect applicability of this hypothesis is furnished by the fact that, as a rule, the values of the ratio $\frac{v}{v^1}$ instead of being constant, vary, sometimes quite materially, with the variation of the number of revolutions. This should not happen, because the velocity of the wake current should vary only with the speed of the ship and should, as a matter of fact, be independent of the number of revolutions of the propellers.

We are acquainted, on the other hand, with the interesting experiments made by Calvert for the purpose of measuring the velocity of the water at the stern in the region in which the propellers work. The results of these experiments, which prove that the velocity of the wake current varies from one stratum to the other are summed up in the two figures of Plate IV, taken from Bertin's "Machines Marines." In these figures are shown approximately the curves of separation of the liquid strata of speeds comprised between $0.5v$ and $0.05v$ in a single-screw ship and in one equipped with twin screws. These curves give us, in the

two cases, a good representation of the variability of the velocity of the wake current in the region of action of the screws.

In the experiments relating to the direction of rotation of propellers Froude has considered it necessary to take into account the influence which might be exerted upon the torsional force by the variability of the wake current, since this influence may differ in the two cases (rotation inboard or outboard), and might attain sufficient importance to cause the propulsion efficiency to vary materially. It being moreover a question of ascertaining differences of small amounts, it becomes necessary to determine the effects of the wake more exactly. I will sum up what constitutes the innovation adopted by Froude, to this end, in the methods of computation which usually follow determinations by experiments.

Let v, n, s, t be respectively the speed, number of revolutions, thrust, and force of torsion in an experiment made with a hull model in conjunction with propeller models. We will now suppose that experiments are made with the same propellers, but by themselves, and dragged a speed v^1, $\frac{v}{v^1}$ being the well known wake coefficient.

From what has been said above it follows that, if the revolutions are the same, n, attained in the preceding experiment, the thrust will likewise be the same, s, and the torsional force *would* also be the same, t, if the wake consisted of parallel particles of water of uniform speed, as has been stated above. But owing to the variable composition of the wake current, the torsional force may be slightly different; let it be t^1, and let the ratio be $\frac{t^1}{t} = \gamma$.

Assuming the hypothesis of uniformity of wake to be correct, t^1 could be made equal to t, or $\gamma = 1$; but this value 1 can no longer be assumed when it is desired to take into account the true action of the wake, and must be determined in each case by experiments.

In other words, the value of γ, the ratio between the torsional force with propellers by themselves and the torsional force with propellers in conjunction with the hull, represents a new factor of

efficiency, a factor which represents the effect of the wake in modifying the efficiency by reason of its turbulence, of its variable velocity from one stratum to another, etc., in the same manner in which the ordinary wake coefficient $\frac{v}{v^1}$ represents the effect of the wake in augmenting the efficiency by reason of its forward movement. This value of γ may also be considered as the ratio between the efficiency of the propellers by themselves and that of the propellers in conjunction with the hull, when the propellers, in the two cases, move at the same rate of travel relative to the water in which they work and at the same number of revolutions.

On the basis of the foregoing it is clear that the propulsive efficiency may be expressed by the formula

$$E = \frac{rv}{t2\pi Rn} = \frac{r}{s} \cdot \frac{sv^1}{2\pi Rt^1 n} \cdot \frac{v}{v^1} \cdot \frac{t}{t^1}.$$

And if we indicate by m and m^1 the turning moments Rt and Rt^1 we shall have

$$E = \frac{r}{s} \cdot \frac{sv^1}{2\pi nm^1} \cdot \frac{v}{v^1} \frac{t^1}{t}$$

Hence the value of efficiency will be obtained from the product of four distinct factors, namely:

1. $\frac{r}{s}$ augment of resistance, or thrust deduction;

2. $\frac{sv}{2nm}$, efficiency of the propellers by themselves at a speed v^1 and at a number of revolutions n.

3. $\frac{v}{v^1}$, coefficient of wake.

4. $\frac{m^1}{m} = \frac{t^1}{t} = \gamma$, factor of influence of the variability of the wake.

In the experiments made by Froude to ascertain the effect of the direction of rotation upon the efficiency, the values of m were generally found to be greater than unity in the case of propellers turning inboard, and smaller than unity in the case of propellers turning outboard. In the former case unity was ex-

ceeded by 0.015 as a maximum; in the latter case unity fell short of being attained by 0.012 as a maximum.

I deem it unnecessary to set forth further details concerning Froude's experiments. I will therefore confine myself to relating the final conclusions to which his experiments have led.

Causing the propellers to turn inboard instead of outboard would give rise—

a. To a slight gain in hull efficiency, due entirely to an increase of the wake coefficient, which increase is compensated in part by a small increase in the coefficient of thrust deduction.

b. To a slight gain in the factor of influence of the turbulence of the wake, which factor is almost invariably larger than unity in the case of propellers turning inboard and smaller in the case of propellers turning outboard.

As mean values Froude has deduced—

Gain in hull efficiency 0.37 per cent.

Gain in the value of γ 1.12 per cent., and consequently a net mean gain of 1.49 per cent. by arranging the propellers to turn inward instead of outward.

The computation of propulsive efficiency according to the original method adopted by Froude, prior to the recent experiments of which I have spoken, was made in substance by determining separately:

1. The value of the coefficient $\frac{r}{s}$.

2. The value of the coefficient $\frac{v}{v^1}$.

3. The value of the efficiency of the screws by themselves corresponding to the point of propulsion of the model.

The values of the first two coefficients were determined by experiments of the hull model in combination with models of propellers; the third value, by experiments with propeller models alone. This method rendered it naturally unnecessary to dedetermine the force of torsion in the experiments with hull and propellers combined.

With the new method, employed as stated, to take account of the effect due to the true nature of the wake current, the determination of the torsional force, in conjunction with the hull model, is necessary. By this method Froude has followed a course which (leaving out of the question some reasons inherent in the exactness of the experimental operations) would have seemed the most natural from the outset. Froude, however, has evidently had reason to bear in mind that the lesser precision of these operations could not lead to much smaller errors than those to which the hypothesis of uniformity of wake current may lead, and has, therefore, recognized the necessity of introducing into the calculation of the efficiency the value of the effective torsional force determined with the propellers working in conjunction with the hull.

This method has, as a matter of fact, long been followed at our experimental tank at Spezia in determining propulsive efficiency.

It will be observed, moreover, that the complete formula of efficiency

$$E = \frac{r}{s} \cdot \frac{sv^1}{2\pi R t^1 n} \cdot \frac{v}{v^1} \cdot \frac{t^1}{t},$$

given by Froude, may be easily reduced to the formula

$$E = \frac{rv}{2\pi R t n} = \frac{r}{\frac{2\pi R}{\frac{v}{n}} \cdot t} = \frac{r}{f},$$

in which the efficiency is given as the ratio between the resistance to towing of the model by itself and the rotative force corresponding to the propulsion of the model as determined with the propellers working in conjunction with the hull.

Returning to Froude's experiments relative to the influence of the direction of rotation of propellers upon the efficiency of propulsion, it will be noted that these experiments were made with the different models of ships used at various rates of speed. It would seem, however, that the ratio between the values of efficiency attained by changing the direction of rotation of the propellers is found to be independent of the speed.

It is not stated what was the position of the propellers relative to the hull in the different experiments made by Froude. Such position, though it may vary only between comparatively moderate limits, is, in my opinion, not without importance. Since it should be remembered that the variations of efficiency observed in changing the direction of rotation of the propellers depend mainly upon the differing actions of the wake current, and since the characteristics of the shape and velocity of such current differ according to the distance from the hull, it is natural to suppose that the effect of changing the direction of rotation may vary according to the different positions occupied by the propellers relative to the hull.

In the discussion which followed the reading of Mr. Froude's interesting paper some cases were cited in which merchant vessels had attained considerable gain in speed by causing the propellers to turn outboard instead of inboard. This result, which is in open contradiction to the results of Froude's experiments, might be due to the particular position of the propellers relative to the hull, as well as to the particular shapes of the sterns of those ships. The latter reason was mentioned by Mr. Froude himself, but it is not precluded that the same importance should be attributed to the other reason.

I will add that in tests made at the model tank in Spezia with a certain number of models of ships and propellers, the influence of the position of the propeller was well demonstrated in some cases. These tests would lead to results rather different from those arrived at by Froude in the majority of cases. This goes to show that, as Froude himself has acknowledged, the experiments made at the Haslar tank are not sufficient to establish a general rule, and that, since cases may occur in which no gain in efficiency would be achieved by causing the propellers to turn inboard, but rather a loss, the best system will always be to ascertain for each individual ship the influence which the change of direction of rotation of propellers will exert upon the value of the propulsive efficiency.

At the Spezia tank, experiments of propulsion were made with two models of battleships, two models of armored cruisers, one model of a protected cruiser and one model of a transport.

In the following table are shown the principal elements relating to the six models subjected to experiments:

Principal data.*	Battleships.		Armored cruisers.		Protected cruiser.	Transport.
	No. 1.	No. 2.	No. 1.	No. 2.		
Length b. p.	44.20	43.33	39.98	41.98	43.80	40.0
Maximum breadth	7.466	7.958	6.70	7.49	4.65	7.02
Mean draught	2.623	2.680	2.27	2.84	1.70	2.70
Difference in draught	0.306	0.00	0.266	0.168	0.150	0.35
Displacement	453.700	469.40	306.83	463.0	174.70	452.12
Wetted surface	371.62	370.50	281.88	367.12	230.0	354.57

*Linear dimensions are in decimeters, areas in square decimeters, and cubical contents (displacement) in cubic decimeters, or liters. R. E. C.

The corresponding data relating to the ships represented by the models, each on a definite scale, are as follows:

Principal data.*	Battleships.		Armored cruisers.		Protected cruiser.	Transport.
	No. 1.	No. 2.	No. 1.	No. 2.		
Length b. p.	132.60	130.00	111.96	104.95	87.60	80.00
Maximum breadth	22.40	23.874	20.12	18.72	9.30	14.04
Mean draught	7.87	8.040	6.82	7.10	3.40	5.40
Difference in draught	0.92	0	0.80	0.42	0.30	0.70
Displacement	12568	13000	8525	7422	1434	3711
Wetted surface	3342	3334	2537	2294	920	1418

*In this table the units are meters for linear dimensions, square meters for areas, and metric tons for displacement.

The hull models were combined with screw models. The screws used in the experiments were of three different types, having the following characteristics respectively:

Type of Screws.	l	l^1	s_5	l	l^1	s_5
Number of blades	3	3	3	.....	.....	.....
	mm.	mm.	mm.	inches.	inches.	inches.
Maximum diameter	190	183.33	170	7.48	7.213	6.69
Pitch, middle of blade	198.55	207.16	229.5	7.812	8.1564	9.04
	mm^2	mm^2	mm^2	sq. in.	sq. in.	sq. in.
Projected area	6821	6182.22	5528	10.573	9 582	8.568
Developed area	8550	7960.55	7150	13.2525	12.339	11.0825
Ratio P ÷ D	1.045	1.13	1.35	.....	.....	.....
Fraction of pitch	0.257	0.251	0.258	.....	.....	.....

These propellers (Plates 1, 2, and 3) are of uniform pitch from the propeller boss (hub) to the periphery and along the generatrices, inclined 16 degrees toward the stern in the case of the "l" and "l^1" propellers and 10 degrees in the case of the "s_5" propellers. The "l" and "l^1" propellers are entirely similar except as to dimensions and the way the blades are fixed to the boss. The "s_5" propellers differ from the "l" propellers in the shape of the blades and in their dimensions.

The hull model of the Battleship No. 1 was combined with "l" propellers, and the latter were arranged in three different positions relative to the hull. The same model was combined with "l^1" propellers, and the latter were tested in a single position relative to the hull.

In conjunction with the model of Battleship No. 2 "l" propellers were tested, and then "l^1" propellers, placed in the same position as the "l" propellers.

The model of Armored Cruiser No. 1 was combined and tried only with "s_5" propellers, but placed in two different positions relative to the hull.

The hull models of Armored Cruiser No. 2 and of Protected Cruiser No. 1 were combined with "l" propellers, which were tested in a single position in the case of each ship.

Finally, the hull model of the transport was tested in conjunction with "s_5" propellers, likewise in a single position relative to the hull.

Dimensions and principal data.	Battleships.				Cruisers.			Transport No. 1.
	No. 1.		No. 2.		Armored.		Prot. No. 1.	
					No. 1.	No. 2.		
	Screw		Screw		Screw	Screw	Screw	Screw
	1	1^1	1	1^1	s_5	1^1	1^1	s_5
No. of screws	2	2	2	2	2	2	2	2
No. of blades	3	3	3	3	3	3	3	3
Max. diameter, m	5.70	5.50	5.70	5.50	5.10	4.58	3.66	3.40
Pitch, middle blade, m.	5.95	6.21	5.95	6.21	6.88	5.18	4.14	4.59
Projected area, m^2	6.14	5.56	6.14	5.56	4.97	3.86	2.47	2.21
Developed area, m^2	7.69	7.16	7.69	7.16	6.43	4.97	3.16	2.86
Fraction of pitch	0.257	0.251	0.257	0.251	0.258	0.251	0.251	0.258
Ratio P/D	1.045	1.13	1.045	1.13	1.35	1.13	1.13	1.35

In the table on page 1091 are shown the dimensions and other principal data which, taking into account the scale of the models, would result for the propellers of the actual ships.

I will now give an account of the different experiments and of the results arrived at through the same.

Experiments with Battleships No. 1, and "I" propellers.—In these experiments the propellers were tested, as stated above, in different positions relative to the hull.

A first series of experiments was made in which the propellers were placed as far forward as possible—that is to say, as far as their diameter and the horizontal distance of the axes from the hull would permit. In a second series of experiments the propellers were moved along the axes (the divergence of which was left unchanged) as far aft as possible. Finally, a third series was made in which the longitudinal position was left as in the second series, but the divergence of the axes was reduced, so that the propellers were brought closer to the diametral longitudinal (midship) plane of the hull.

The position of the propellers relative to the hull, in the three series of experiments, is shown in the following table, in which the different distances refer to ships of actual size.

	Series I.	*Series II.*	*Series III.*
Distance of the center of the screw bosses from the line $P_P A_D$, meters	6.87	3.37	3.37
Distance of the center of the screw bosses from the diametral (midship) plane, meters	4.32	4.42	3.80
Distances of the center of the screw bosses from the water line, meters	5.20	5.20	5.20
Minimum distance of the tips of the blades from the hull, meters	0.20	1.00	0.46

The experiments were executed at a speed of about 2.022 meters a second, corresponding to 21.5 knots in the case of actual ships; but as it was naturally not practicable to maintain the same speed in all the experiments, the final results relate to somewhat different rates of speed varying by a few tenths of a knot from the rate of speed chosen.

The model with which the experiments were made was fitted

with brackets and propeller-bearing shafts; but as the other models, the experiments with which will be described later on, did not have such fittings, I will, for the sake of uniformity, give the values of the different coefficients irrespective of fittings—that is to say, considering the resistances of the model equipped with the fittings referred to as the resistances of the bare model.

In the following table are summed up the principal results of the three series of experiments or the value of the elements.

v = velocity of the model in meters per second.
r = resistance of the model to towing in kilograms.
s = thrust of the propellers corresponding to the propulsion, in kilograms.
t = torsional force employed on the propeller models and corresponding to s, in kilograms.
t^1 = torsional force employed on the propellers by themselves at a speed v' and corresponding to the simultaneous values s, n, in kilograms.
n = number of revolutions corresponding to the propulsion.
$\frac{r}{s}$ = coefficient of augment of resistance, or thrust deduction.
$\frac{v}{v^1}$ = coefficient of wake.
$\frac{r}{s} \cdot \frac{v}{v^1}$ = hull efficiency.
$\gamma = \frac{t^1}{t}$ = coefficient characteristic of the wake.

Experimental determinations.	v	r	s	n	$\frac{r}{s}$	$\frac{v}{v^1}$	$\frac{r}{s}\frac{v}{v^1}$	t	t^1	γ
1. Propellers turning inboard...	2.037	3.855	4.470	12.06	0.863	1.070	0.923	1.974	1.820	0.923
outboard.	2.034	3.480	4.380	11.84	0,877	1.086	0.951	1.890	1.768	0.935
2. Propellers turning inboard...	2.024	3.725	4.190	11.87	0.888	1.064	0.946	1.809	1.738	0.952
outboard.	2.024	3.740	4.165	11.76	0.898	1.081	0.971	1 820	1.718	0.943
3. Propellers turning inboard...	2.041	3.780	4.350	11.99	0.870	1.070	0.931	1.872	1.790	0.955
outboard.	2.045	3.800	4.365	11.71	0.871	1.111	0 968	1.827	1.758	0.962

The final results, as relating to an actual ship, are therefore as follows:

Results.	Series I. Screws turning In	Series I. Screws turning Out.	Series II. Screws turning In	Series II. Screws turning Out	Series III Screws turning In	Series III Screws turning Out
Speed in knots	21.69	21.66	21 55	21 60	21.73	21 77
No. of revolutions per minute	132.1	129.9	130.1	128.9	131.4	128.4
Total propulsive efficiency	0.509	0.537	0.541	0.543	0.528	0.558

The results above reported show:

1. That with the propellers arranged in the second position (far aft and well removed from the hull) the efficiency is practically the same, whether the propellers turn outboard or toward the hull.

2. That with the propellers arranged in the second position, but much closer to the hull, third position, there is a non-negligible loss of efficiency when the propellers turn inboard. The efficiency with the propellers turning outboard is somewhat higher than in the second position.

3. That with the propellers arranged in the first position (well forward and very close to the hull) there is likewise considerable loss in efficiency when the propellers turn inboard. The efficiency with the propellers turning outboard is less than that corresponding to the second and third positions.

4. That the loss in efficiency encountered in the case of inboard-turning propellers is due to a decrease of the coefficient of thrust deduction, or to greater disturbance produced by the propellers, to a decrease of the wake coefficient or less good utilization of the wake, and to a decrease of the coefficient γ.

This last coefficient proves to be higher for propellers turning inboard only in the case corresponding to the second position.

Experiments with Battleship No. 1 and "1^1" propellers.—The experiments were made with the propellers in the third position. The speed was practically the same as in the preceding experiments.

The results are summed up in the following table:

Experimental determinations	v	r	s	n	$\frac{r}{s}$	$\frac{v}{v^1}$	$\frac{r}{s}\frac{v}{v^1}$	t	t^1	γ
Screws turn inboard	2.036	3 710	4 280	12.41	0.867	1.099	0.951	1 818	1.796	0 989
outboard	2.044	3.744	4.160	12.21	0.900	1.122	1.010	1.748	1.755	0.996

The final results as relating to an actual ship are shown in the following table:

Results.	Screws turning	
	inboard.	outboard.
Speed in knots	21.68	21.76
Revolutions per minute	127.10	125.4
Total efficiency	.554	.581

These results entirely confirm those obtained in the preceding experiments with the propellers in the third position. The difference in the values of efficiency, however, proved to be slightly less in the former experiments.

Experiments with Battleship No. 2 and "l" propellers.—The "l" propellers of 5.7 meters (diameter) were tested in a single position relative to the hull and at a single speed. The "l^1" propellers of 5.5 meters were likewise tested at a single speed and in the same longitudinal position as the "l" propellers.

The position of the "l" propellers relative to the ship in these experiments is indicated by the following data:

Distance of the center of the bosses from the ordinate $P_p A_D$, meters, 3.300
Distance of the center of the bosses from the longitudinal diametral plane, meters..... 3.800
Distance of the center of the bosses from the water line, meters..... 4.790
Minimum distance of the tips or the blades from the hull, meters..... 0.700

The results of the experiments are shown in the following table:

Experimental determinations.	v	r	s	n	$\frac{r}{s}$	$\frac{v}{v^1}$	$\frac{r}{s}\frac{v}{v^1}$	t	t^A	γ
Propellers turn'g inboard	1.667	2.690	3.040	10.35	0.886	1.090	0.965	1.305	1.288	0.987
outboard	1.679	2.735	3.090	10.22	0.886	1.129	1.000	1.332	1.315	0.988

For the actual ship we obtain the following final results:

Results.	Screws turning	
	inboard.	outboard.
Speeds in knots	17.74	17.80
Revolutions per minute	106.8	105.8
Total efficiency	0.540	0.548

These results show that in the case corresponding to the experiments the propulsive efficiency with the propellers turning inboard is slightly lower than that obtained with the propellers turning outboard.

In the second series of experiments with the same hull, but with "l^1" propellers of a diameter corresponding to 5.5 m. in case of the actual ship, the experimental investigation was made at a higher speed than in the preceding experiments. Such speed corresponds to the maximum speed provided for the actual ship, namely, about 20 knots. The propellers were arranged with the centers of the bosses in the same position as in the case of the "l" screws, of a diameter corresponding to 5.7 meters.

The results obtained are as follows:

Experimental determinations.	v	r	s	n	$\frac{r}{s}$	$\frac{v}{v^1}$	$\frac{r}{s}\frac{v}{v^1}$	t	t^1	γ
Propellers turn inboard	1.880	3.080	3.535	11.29	0.871	1.098	0.956	1.490	1.470	0.987
outboard	1.876	3.060	3.440	11 26	0.889	1.098	0.976	1.470	1.461	0.994

The final results relating to the actual ship are as follows:

Results.	Screws turning	
	inboard.	outboard.
Speed in knots	20.01	19.97
Revolutions per minute	116.3	115.9
Total efficiency	.558	.563

Experiments with Armored Cruiser No. 1 and "l^1" propellers.—With the Armored Cruiser No. 1 the experiments were made with the propellers in two different positions relative to the hull. These

positions give the following results, which relate to the actual ship:

	First position.	Second position.
Distance of center of bosses from the ordinate $P_p A_D$, meters	0	0
Distance of the center of bosses from the longitudinal plane, meters	4.70	3.072
Distance of center of bosses from the water line, meters	4.00	4.00
Minimum distance of tips of blades from the hull, meters	2.20	0.350

The results of the experiments are shown in the following table:

Experimental determinations.	v	r	s	n	$\frac{r}{s}$	$\frac{v}{v^1}$	$\frac{r}{s}\frac{v}{v^1}$	l	l^1	γ
1st position, Propellers turn inboard	1.814	2.170	2.320	10.30	0.937	1.044	0.978	1.242	1.228	0.986
outboard	1.868	2.200	2.355	10.40	0.937	1.042	0.976	1.252	1.246	0.992
2d position, Propellers turn inboard	1.855	2.170	2.355	10.14	0.922	1.081	0.997	1.224	1.212	0.989
outboard	1.855	2.170	2.335	10.13	0.925	1.081	1.000	1.217	1.208	0.992

The final results for the actual ship are as follows:

Results.	Turning inboard.		Turning outboard.	
	First position.	Second position.	First position.	Second position.
Velocity in knots	19.71	19.72	19.86	19.72
Revolutions per minute	106.2	104.08	106.7	103.75
Total efficiency	0.512	0.528	0.512	0.533

It should be stated that in both positions the centers of the propeller bosses corresponded to the line $P_p A_D$. This position, which could not be adopted in practice owing to the rudder, was chosen only to render it possible, in view of the shape of the hull, to bring the axes considerably closer to the longitudinal diametral plane. In the first position the axes were far removed from the hull—to a degree which would probably be impracticable in the case of the actual ship—the object being to place the propellers as far as possible from the disturbing action of the hull. As may be seen from the results, this position resulted in considerably

less efficiency owing to a poorer utilization of the wake current, since the propellers, at this great distance from the hull, worked, no doubt, in a region in which the wake has considerably less speed than was the case in close proximity to the hull. This removal of the axes away from the hull, which in the preceding experiments had reduced the difference in efficiency encountered upon changing the direction of rotation of the propellers, in these latter experiments entirely obliterated this difference, which, while not great, was nevertheless noticeable with the propellers in the second position, and always in favor of the outboard-turning propellers.

Experiments with Armored Cruiser No. 2 and "I¹" propellers.—The propellers were arranged, relative to the hull, in the position indicated by the following distances, which relate to the actual ship:

Distance of the center of the bosses from the ordinate $P_p A_D$, meters ... 2.50
Distance of the center of the bosses from the longitudinal plane, meters, 2.95
Distance of the center of the bosses from the water line, meters..... 4.35
Minimum distance of the tips of the blades from the hull, meters..... 0.50

The experiments were made at a speed of 1.85 meters a second, corresponding to 18 knots in the case of the actual ship, and the following results were obtained:

Experimental determinations.	v	r	s	n	$\frac{r}{s}$	$\frac{v}{v^1}$	$\frac{r}{s}\frac{v}{v^2}$	t	t^1	γ
Screws turn inboard.....	1.853	3.095	3.530	11.33	0.877	1.091	0.956	1.485	1.468	0.987
outboard	1.830	3.080	3.435	11.26	0.896	1.092	0.979	1.460	1.448	0.991

The final results for the actual ship are as follows:

Results.	Screws turning	
	inboard.	outboard.
Speed in knots.....	18.01	17.98
Revolutions per minute.....	128.8	127.68
Total efficiency.....	0.557	0.565

In this case, also, the difference in efficiency due to the changed direction of rotation of the propellers, while small, is in favor of the rotation outward.

Experiments with Protected Cruiser No. 1 and "l^1" propellers.—The position of the propellers relative to the hull is shown by the following distances, which, as in preceding cases, are for a full-sized ship:

Distance of center of bosses from the ordinate P_p A_D, meters..... 2.84
Distance of center of bosses from the longitudinal plane, meters 2.20
Distance of center of bosses from the water line, meters..... 2.76
Minimum distance of the tips of the blades from the hull, meters..... 0.30

The experiments, in this case, were made at a speed of about 2.42 meters a second, corresponding to about 21 knots for the actual ship. The results of the experiments and the final results relating to the actual ship are shown in the following two tables, respectively:

Experimental determinations.	v	r	s	n	$\frac{r}{s}$	$\frac{v}{v^1}$	$\frac{r}{s}\frac{v}{v^1}$	t	t^1	γ
Screws turn inboard.....	2.424	3.185	3.420	13.62	0.932	1.030	0.960	1.692	1.660	0.980
outboard.....	2.422	3.170	3.375	13.59	0.939	1.030	0.967	1.670	1.640	0.981

Results.	Screws turning	
	inboard.	outboard.
Speed in knots	21.07	21.05
Revolutions per minute.....	175.9	175.3
Total efficiency.....	0.546	0.552

A slight gain in efficiency is achieved in this case also by causing the propellers to turn outward.

Experiments with Transport No. 1 and "s_8" propellers.—The position of the propellers relative to the hull is indicated by the following distances:

Distance of center of bosses from ordinate P_p A_D, meters..... 1.30
Distance of center of the bosses from the longitudinal diametral plane, meters 2.30
Distance of the center of the bosses from the water line, meters..... 2.76
Minimum distance of the tips of the blades from the hull, meters..... 0.50

The experiments were made at a speed of about 1.30 meters a second, corresponding to 12 knots for the actual ship. The direct results of the experiments and the final results for the actual ship are shown in the two tables following, respectively:

Experimental determinations.	v	r	s	n	$\frac{r}{s}$	$\frac{v}{v^1}$	$\frac{r}{s}\frac{v}{v^1}$	t	t^1	γ
Screws turn inboard	1.396	1 680	1.960	7.83	0.857	1.201	1.030	0.885	0.892	1.008
outboard	1.385	1.640	1.865	7.75	0.880	1.191	1.049	0.885	0.867	1.013

Results.	Screws.	
	Turning inboard.	Turning outboard.
Speeds in knots	12.13	12.04
Revolutions per minute	97.56	96.48
Total efficiency	0.550	0.564

In this case the gain in efficiency achieved by causing the screws to turn outward is quite considerable.

The results of the experimental findings used in the investigations which I have related were in each case subjected to the most rigorous verification. The most scrupulous care was always observed in order to obviate the slightest causes of error in the course of the experiments. In some instances the experiments were repeated by different operators, and in each case the accuracy of the first observation was confirmed.

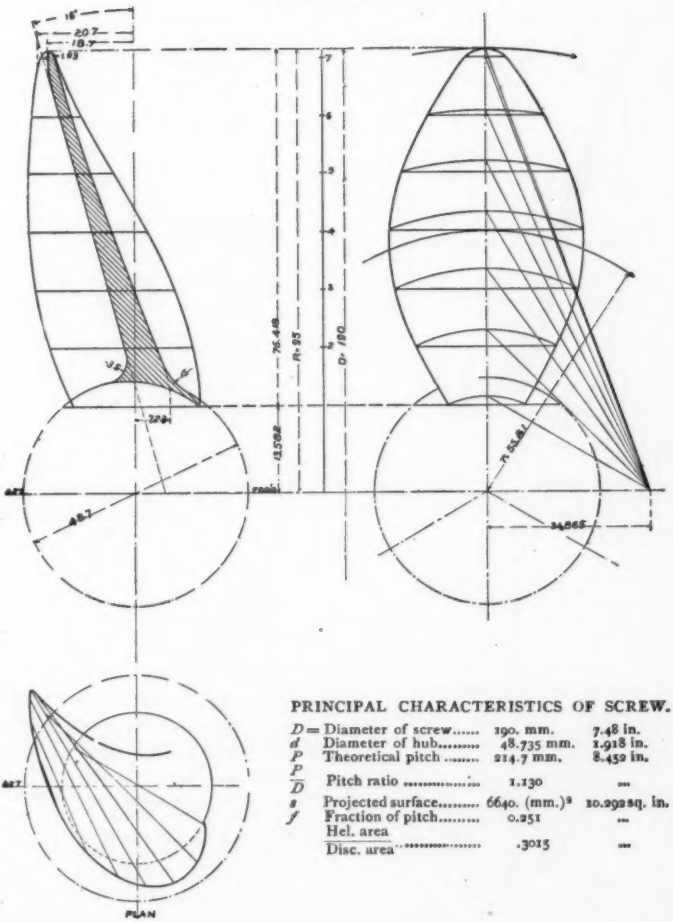
In order to prove the always perfect agreement of the results and the care with which the experiments were carried on, there are shown in Plate V the diagrams which relate to the propulsion experiments made with the model of Battleship No. 1 in conjunction with models of "l^1" propellers. These experiments, made the first time on April 26, were repeated on May 14 by a different operator, after the water in the tank had been changed.* The practical agreement of the results obtained in the two cases will appear from an examination of the diagrams.

*The water in the tank is changed when it is deemed that it is no longer sufficiently limpid—usually in May and November of each year.

DIRECTION OF ROTATION OF SCREW

PLATE I.

SCREW TYPE 'I'



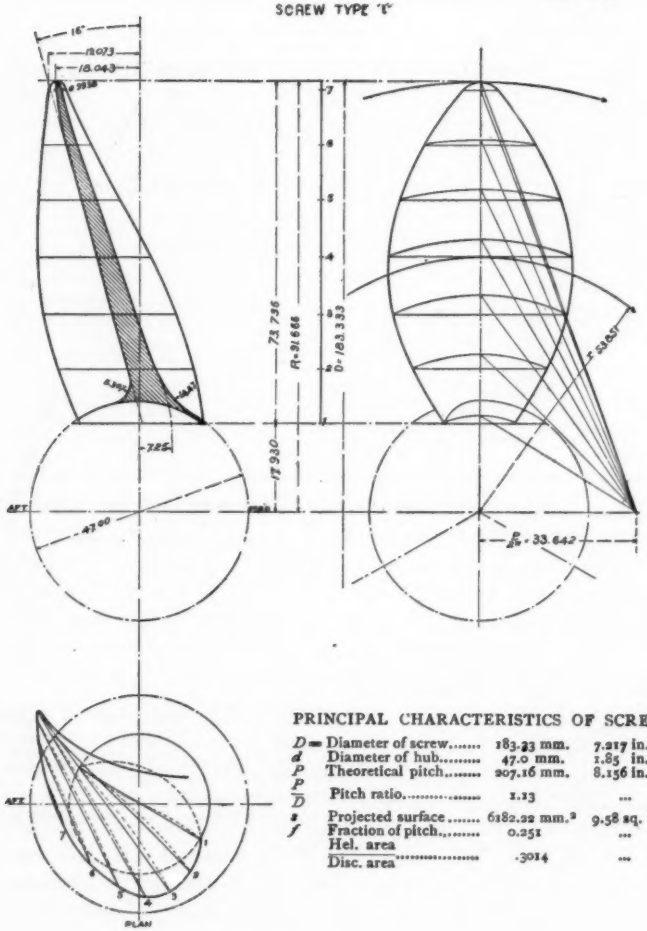
PRINCIPAL CHARACTERISTICS OF SCREW.

D =	Diameter of screw.....	190. mm.	7.48 in.
d	Diameter of hub.....	48.735 mm.	1.918 in.
P	Theoretical pitch	214.7 mm.	8.452 in.
$\frac{P}{D}$	Pitch ratio	1.130	...
s	Projected surface.....	6640. $(mm.)^2$	10.292 sq. in.
f	Fraction of pitch.....	0.251	...
	Hel. area / Disc. area	.3015	...

DIRECTION OF ROTATION OF SCREW

PLATE II

SCREW TYPE 'E'



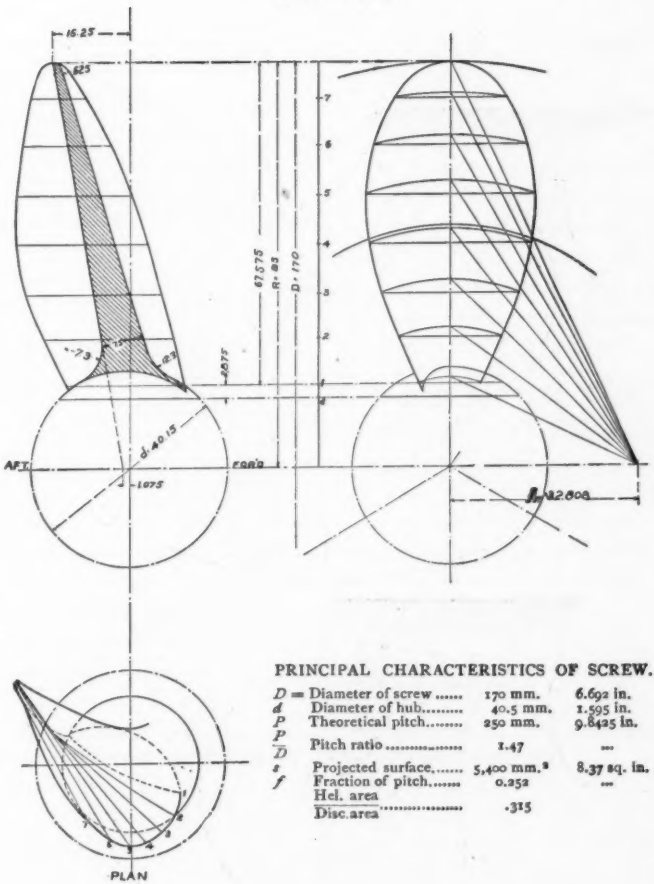
PRINCIPAL CHARACTERISTICS OF SCREW.

D	Diameter of screw.....	183.33 mm.	7.217 in.
d	Diameter of hub.....	47.0 mm.	1.85 in.
P	Theoretical pitch.....	207.16 mm.	8.156 in.
$\frac{P}{D}$	Pitch ratio.....	1.13	...
s	Projected surface.....	6182.22 mm.^{2}	9.58 sq. in.
f	Fraction of pitch..... Hel. area / Disc. area	0.251 .3014	... ...

DIRECTION OF ROTATION OF SCREW

PLATE III.

SCREW TYPE $\frac{6}{5}$



PRINCIPAL CHARACTERISTICS OF SCREW.

D =	Diameter of screw	170 mm.	6.692 in.
d	Diameter of hub.....	40.5 mm.	1.595 in.
P	Theoretical pitch.....	250 mm.	9.8425 in.
$\frac{P}{D}$	Pitch ratio	1.47	...
s	Projected surface.....	5,400 mm.^{2}	8.37 sq. in.
f	Fraction of pitch.....	0.252	...
	$\frac{\text{Hel. area}}{\text{Disc area}}$	.315	

DIRECTION OF ROTATION OF SCREW

PLATE IV.

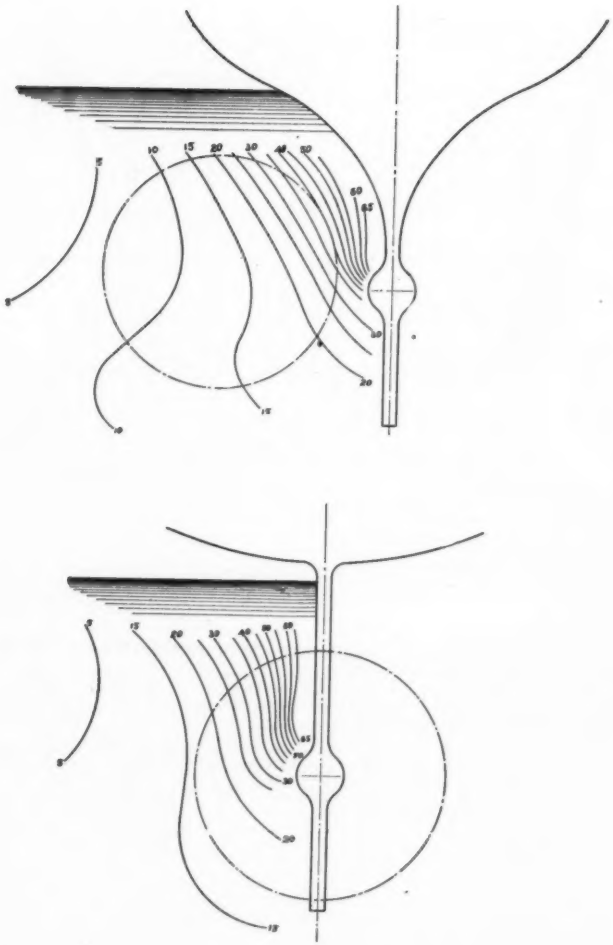
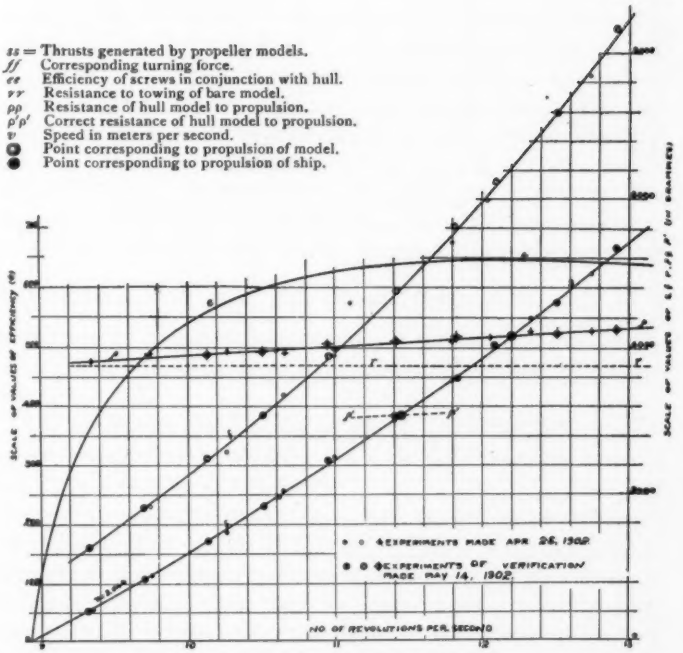
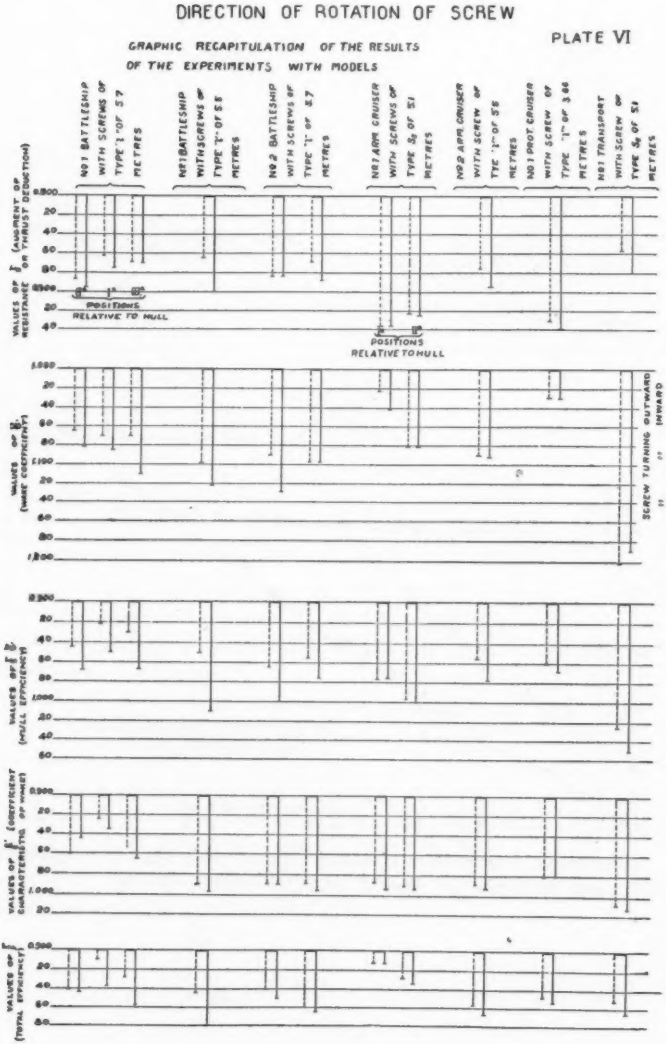


PLATE V.


ss = Thrusts generated by propeller models.
ff Corresponding turning force.
ee Efficiency of screws in conjunction with hull.
vv Resistance to towing of bare model.
ρρ Resistance of hull model to propulsion.
ρ'ρ' Correct resistance of hull model to propulsion.
v Speed in meters per second.
Point corresponding to propulsion of model.
Point corresponding to propulsion of ship.
EXPERIMENTS MADE APR. 26, 1902
EXPERIMENTS OF VERIFICATION MADE MAY 14, 1902
NO. OF REVOLUTIONS PER SECOND

Experiments of propulsion with model of Battleship No. 1 in conjunction with propellers of the "*l*[1]" type of 183.33 millimeters diameter.


DIRECTION OF ROTATION OF SCREW
PLATE VI
GRAPHIC RECAPITULATION OF THE RESULTS
OF THE EXPERIMENTS WITH MODELS
No 1 BATTLESHIP WITH SCREWS OF TYPE "I" OF 5·7 METRES
No 1 BATTLESHIP WITH SCREWS OF TYPE "I" OF 5·8 METRES
No 2 BATTLESHIP WITH SCREWS OF TYPE "I" OF 5·7 METRES
No 1 ARM CRUISER WITH SCREWS OF TYPE S_2 OF 5·1 METRES
No 2 ARM CRUISER WITH SCREW OF TYPE "I" OF 5·8 METRES
No 1 PROT CRUISER WITH SCREW OF TYPE "I" OF 3·66 METRES
No 1 TRANSPORT WITH SCREW OF TYPE S_2 OF 5·1 METRES
VALUES OF $\frac{T}{R}$ (RATIO OF THRUST TO RESISTANCE)
POSITIONS RELATIVE TO HULL
VALUES OF $\frac{V_a}{V}$ (WAKE COEFFICIENT)
SCREW TURNING OUTWARD
INWARD
VALUES OF $\frac{R}{T}$ (HULL EFFICIENCY)
VALUES OF $\frac{V_a}{V}$ (COEFFICIENT CHARACTERISTIC OF WAKE)
VALUES OF $\frac{R}{P}$ (TOTAL EFFICIENCY)

It has been deemed convenient to unite in a graphic table the principal factors of propulsive efficiency with reference to the different experiments made, and in this table, which permits an easier and quicker comprehension of the results obtained, I have followed a system quite similar to the one employed by Froude. In Plate VI the different factors of efficiency are represented by segments, composed of continuous lines in the case of outboard-turning propellers, and of dotted lines for inboard-turning propellers. In the first diagram are given the values of the coefficient of thrust deduction; in the second, the values of the wake coefficient, or more correctly speaking, the amounts by which such coefficient exceeds unity in the various cases; in the third, the values of the hull coefficient—the amounts by which such coefficient exceeds unity being noted above the horizontal axis, those by which such coefficient falls short of unity, below the horizontal axis; in the fourth diagram are represented the values of the coefficient $\frac{\ell}{\ell^1}$ the values in excess of unity being noted above the axis, those inferior to unity below. Finally, the values of the "total propulsive efficiency" are shown in the fifth diagram.

The conclusions which may be drawn from the results of the experiments with the models of the six different types of ships may be summed up as follows:

1. As regards Battleship No. 1, with propellers placed at the normal distance from the hull, or much closer to it, the efficiency attained with the propellers turning inboard is considerably lower than that attained when they are fitted to turn outboard when going ahead. By changing the direction of rotation of the propellers when the same are removed to a considerable distance from the hull, the efficiency is not impaired.

2. In the case of Battleship No. 2, causing the propellers to turn inboard gives rise to a negligible decrease of efficiency. Such decrease remains practically the same with propellers of the same type but somewhat smaller.

3. As regards Armored Cruiser No. 1, when the propellers are removed to some distance from the hull, the efficiency does not

vary upon a change of direction of rotation. A slight decrease in efficiency becomes apparent with inboard-turning propellers when the same are brought very close to the hull.

4. In the case of Armored Cruiser No. 2 and Protected Cruiser No. 1, with propellers in the normal position relative to the hull, the efficiency decreases to a considerable degree in the case of the former, and to a scarcely perceptible degree in the case of the latter, when the propellers turn inboard.

5. As regards the transport (a ship having a very round, full stern) with propellers in the normal position relative to the hull, the efficiency is considerably lower when the propellers turn inboard than that attained with them turning outboard.

The results of the experiments made seem to prove, in some cases, that the position of the propellers affects the variation of efficiency due to the changed direction of their rotation. It seems that the difference between the value of the propulsive efficiency when the propellers turn inboard and that of the efficiency attained with outboard-turning ones is greater the less the propellers are removed from the longitudinal plane of the ship—that is to say, the greater the difference of speed of the particles of water composing the wake current in the region in which the propellers work. And it is perhaps in these differences of speed and in the effect which they exert upon the action of the propellers that we should look for an explanation of the variations in efficiency which accompany the change of direction of rotation. I do not hesitate to state, however, that it would be unwise to draw conclusions of a general character even from a large number of experiments. It will always be the best plan to make experiments of propulsion with the greatest care in the case of each individual ship, so as to determine in each case whether any gain, and, if so, what gain, in efficiency may be expected from causing the propellers to turn in one direction rather than in the other.

TRANSLATOR'S NOTE.—The ordinate $P_p A_D$ referred to in the text, in locating the positions of the screw boss centers, is a line passing through the foot of a perpendicular from that center to a transverse plane through the after perpendicular of the vessel.

In other words, this coördinate simply locates the screw boss center with regard to the plane spoken of, as the other two coördinates locate it with reference to the midship plane of the vessel and its distance below the water line.

It is interesting, in view of the fact that, a year or so ago, a concensus of European naval opinion placed the *Vittorio Emanuele III* type ahead of all existing designs of battleships, as possessing the most admirable combination of speed, armor, battery and radius of action, to note that the model designated Battleship No. 1 is that of this type of ship. According to the designer, Colonel Cuniberti, of the R. I. Navy, in addition to a waterline belt of 10-inch armor with an 8-inch belt for part of the length above that, and a battery comprising two 12-inch guns and twelve 8-inch guns, she is still to have 20,000 I.H.P., capable of giving a speed of 22 knots and all on a displacement of about 12,600 tons.

The part which propeller experiments have had in making part of this possible, is shown in the high propulsive efficiencies achieved by working out and studying the various possibilities, as appears to have been done in this case.

Model Battleship No. 2, similarly represents the *Benedetto Brin* type, while Armored Cruiser No. 2 and Protected Cruiser No. 1 represent the *Garibaldi* and *Agordat* classes respectively.

THE U. S. S. *CLEVELAND.*

GENERAL DESCRIPTION—OFFICIAL TRIAL.

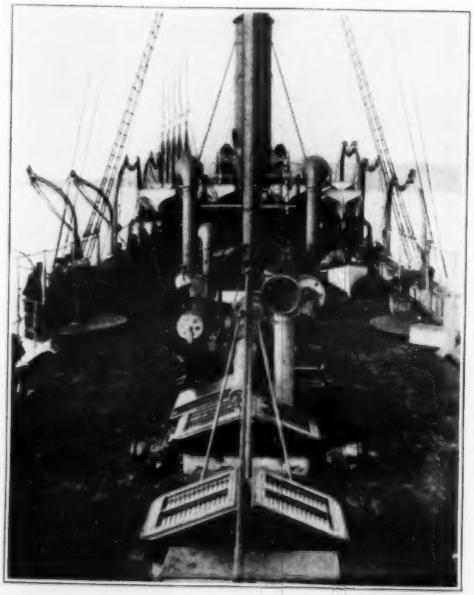
By Lieutenant Commander F. C. Bowers, U. S. N.

The *Cleveland* is a sheathed protected cruiser contracted for by the Bath Iron Works, Bath, Maine, December 14th, 1899, the price being $1,041,650, and the time for completion thirty months.

The keel was laid June 1st, 1900, and the vessel was launched September 21, 1901. The total weight of machinery was limited to 427 tons, including engines, boilers and appurtenances, all fixtures in engine and fire rooms, smoke pipes, distilling apparatus, tools in workshop, heating apparatus, stores, spare parts and water in boilers, condensers, pumps, pipes and stern tubes, but not including reserve feed water, capstans, windlass, steering gear or winches. The penalty forover weight was $500 a ton.

The speed to be developed was 16½ knots an hour for four consecutive hours with an air pressure in the ash pits of one (1) inch of water, the vessel to be weighted to a mean draught corresponding to a trial displacement of 3,200 tons, subject to deductions from the price of the vessel, on account of her failure to reach the speed guaranteed, at the rate of $25,000 a quarter knot for speed between 16½ and 16 knots, and $50,000 a quarter knot for speed between 16 and 15½ knots, but if the vessel failed to exhibit an average speed of at least 15½ knots an hour it was to be optional with the Secretary of the Navy to reject or accept her at a reduced price.

Length between perpendiculars, feet and inches	292- 0
on load water line, feet and inches	292- 0
over all, feet and inches	308- 9⅝
Beam, feet and inches	44- 0



U. S. S. "CLEVELAND."

Ratio of length to beam = 292 ÷ 44 =	6.63
Depth, molded, from top of main deck beam at side to bottom of frame at midship section, feet and inches	30– 3
forward, seagoing trim, normal, feet and inches	15– 2⅜
aft, seagoing trim, normal, feet and inches	16– 0⅜
Draught, mean, seagoing, trim, normal, feet and inches	15– 7½
Displacement for above draught, tons	3,164
per inch immersion, at load water line, tons	22.3
Area immersed midship section, square feet	602
Center of gravity of water line, 3.8 feet aft of frame No. 49, feet and inches	15– 7½
Center of buoyancy above bottom of keel, feet and inches	9– 2⅜
Center of buoyancy aft of midship section, ship on even keel, feet.	.66
Center of gravity above bottom of keel, feet and inches	16– 9
Transverse metacenter above center of buoyancy, feet and inches	10– 2
Coefficient of fineness in extreme dimensions, per cent	55.1
of midship section, per cent	87.5
load water line, per cent	72.8
Cylindrical coefficient, per cent	63

Number of frames, 94; spaced, 3 feet apart.

Number of watertight compartments 148, including 25 cofferdam compartments.

DETAIL OF HULL CONSTRUCTION.

Flat Keel.—Made of two thicknesses, inner course of 15-pound plate, 32 inches wide; outer course 20-pound plate, 42 inches wide.

Vertical Keel.—Thirty-four inches in depth throughout except at ends of 15-pound plate.

Docking Keel.—Of East India teak 18 inches wide and 10 inches deep, bottom covered with 30-ounce copper. Beneath this is a false keel of 2-inch white oak.

Stem.—Lower section to 9 feet above load water line of cast manganese-bronze cast in one piece, upper section of 9-inch by 4-inch bar iron. Weight of casting, 17,470 pounds.

Stern Post.—Of manganese-bronze in one piece, weighing 17,264 pounds.

Rudder.—Not balanced. Frame of manganese-bronze filled in with white pine and covered with ¼-inch manganese-bronze plates.

Transverse Frames.—There are ninety-four frames spaced 36 inches apart. From frame 18 to 86 they are formed of 6-inch by

3½-inch by 3½-inch by 15.5-pound Z bars, with reverse bars 3-inch by 3-inch by 7-pound angles. From 17 forward and 87 aft they are of 4-inch by 3½-inch by 9.1-pound angles with 3-inch by 3-inch by 7-pound angle reverse bars.

Longitudinals.—The first and second longitudinals extend from frame 17 to frame 80 and are of 12½-pound plate. The third, extending from frame 24 to 69, is of 15-pound plate.

Bilge and Side Stringers.—These consist of intercostal 12½-pound plates riveted to a continuous Z bar 6 inches by 3½ inches by 3½ inches by 15.6 pounds per foot, worked on inner edge of frames from frame No. 34 to frame No. 69. The intercostal plates are flanged and riveted to shell.

Deck Beams.—Main, gun and protective-deck beams are of bulb angles 7 inches by 3 inches by 18¼ pounds per foot. Berth deck, angle bars 4½ inches by 3 inches by 9.1 pounds per foot. Magazine and platform: Angle bars 4½ inches by 3 inches by 9.1 pounds per foot.

Bridge Beams.—Angle bars 4½ inches by 3 inches by 9.1 pounds per foot.

Outer Plating.—Of 15-pound plate throughout, except the strake connected to third longitudinal, which is 17½ pounds per foot for the length of the longitudinal and 15 pounds per foot at ends.

Inner Bottom.—Extends from frame 17 to frame 85. From frame 34 to frame 69 the plating is 12½ pounds per square foot. Forward and aft of these frames it is of 10-pound plate. The spaces from frames 41 to 50, inclusive, and from the vertical keel to third longitudinal on both sides of ship, are fitted as reserve feed tanks with a capacity of 40 tons.

Material.—Mild steel, of domestic manufacture, conforming to requirements of "Specifications for Inspection of Hull Material for Vessels of the U. S. Navy," approved by the Secretary of the Navy June 19, 1899.

Coffer Dam.—A coffer dam about 27 inches wide between berth and protective decks extends from the stern post forward to bulkhead No. 7, and is filled with fire-proofed corn-pith cellulose packed in briquettes.

Upper Bridge Deck.—Contains the two 30-inch searchlights.

Bridge Deck.—Contains the steering wheel, binnacle and telegraphs.

Main Deck.—Contains the pilot house and chart room, two 5-inch guns, one foward and one aft, four Colt's automatic guns, four 6-pounders, platform for after searchlights and superstructure for boats.

Gun Deck.—Contains Captain's cabin, executive officer's and navigator's quarters aft, and forward is the sick bay, dispensary and paymaster's office. On this deck are six 5-inch guns and four 6-pounders, two amidship and two in forward peak.

Berth Deck.—On this deck, aft, is the ward-room and warrant-officers' quarters, and forward is the engineers' workshop, ice machine and refrigerating plant, crews' space and prison.

Protective Deck.—The amidship portion of this deck is taken up with boiler and engine-room hatches and coal bunkers, and forward and abaft with store rooms. See Fig. 10 for coal stowage. The flat deck is of 12½-pound plate, and on the slope the lower course is of 20-pound plate. On the slope between frames 34 and 69, upon the 20-pound plate, there is a course of 80-pound nickel-steel armor plate.

Platform Deck.—Extends from frames 34 forward to stem and from 69 aft to stern, and is devoted entirely to stores.

Hold.—The hold is given to engine and boiler rooms, coal bunkers, magazines and stores. Fig. 2 gives a general arrangement of machinery and Figs. 4 and 5 the stowage of coal.

Drainage System.—The main drain is of 10-inch galvanized-iron pipe, and drains the engine and boiler compartments, and is connected to circulating pump, with branch connections to the fire and bilge and auxiliary feed pumps. The 5-inch auxiliary drain drains the engine and boiler compartments, with a 2½-inch branch connection to the inner bottom compartments, and has five manifolds with connections to the auxiliary feed and fire and bilge pumps, and also two connections at berth deck for the 6-inch portable handy-billy. The manifold valves are operated from the manifold direct.

Armament.—

Ten 5-inch guns, 50 calibers in length.

Two 1-pounders,
Eight 6-pound, rapid fire,
Four Colts' automatic guns,
One field gun.
} Secondary battery.

Only the 5-inch guns on gun deck are protected by 1¼-inch nickel-steel. The guns on main deck have no shields.

Boats.—

One 30-foot steam cutter,	. .	weight, 11,861 pounds.
One 30 foot launch,	. . .	weight, 7,038 pounds.
Four 28-foot cutters, each,	.	weight, 4,927 pounds.
One 28-foot gig whaleboat,	.	weight, 4,220 pounds.
One 28-foot whaleboat,	. .	weight, 4,348 pounds.
One 20-foot dinghy,	. . .	weight, 2,080 pounds.
One 16-foot dinghy,	. . .	weight, 1,325 pounds.

False Ceiling.—There is a false ceiling under the protective deck extending throughout the length of the engine and fire rooms. It consists of two layers of 1½-inch fire felt and of one layer of ⅛-inch asbestos mill board secured under deck beams where possible, thus giving an air space under the deck. In many instances it was found impossible to do this, and in such cases the fire felt was fitted in to suit the conditions.

INTERIOR COMMUNICATION.

Mechanical Telegraphs and Gongs.—One gong in each engine room, with pulls on quarter deck and bridge.

One gong at each ammunition hoist, with pulls in handling rooms.

One gong at each anchor engine, with pull on main deck.

One gong at each ash hoist in fire room, with pull at hoisting engine.

There is a repeating mechanical telegraph of Cory & Sons' design for each engine room, connected to transmitters on the bridge. There is also a repeating mechanical telegraph between the engine and fire rooms.

Mechanical tell-tales, of the contractors' design, showing di-

rection and speed of engines, are fitted in the pilot house and on the bridge. In each engine room there is a dial with one pointer which, by means of differential gear, shows the relative speed of the two engines, the pointer turning in the direction of, and at the increased speed of, the faster engine.

Voice Pipes.—There is a system of voice pipes, with call bells, connecting the various parts of the ship. There is also a system of push buttons, connecting the various pantries with the Captain's and ward-room officers' quarters. There are two boat-hour gongs in the warrant and ward-room officers' country, with push buttons on main deck.

Fire Alarms.—There are mechanical thermostats in the coal bunkers, magazines and store rooms, connected to an annunciator on gun deck near executive officers' quarters.

Water-Tight Door Alarms.—There are ten solenoid whistles located on berth deck and in machinery space as signals for closing watertight doors. The contact maker is in the chart house.

General Alarms.—There are eleven general alarm gongs, located on gun and berth decks, with contact makers in chart house, Captain's state room and executive officer's office.

Electric Engine Telegraph.—These are of the illuminated dial pattern, and will signal change from 1 to 4. There is an indicator on each engine, each connected to a combined indicator and transmitter on bridge. The indicators on engines have a magneto bell placed on the back.

Steering Telegraphs.—Of the illuminated dial pattern, and will signal "Steady" and "0," white light; "5," "15" and "35" (degrees) starboard, green light; "5," "15" and "35" (degrees) port, red light. Indicators are located in the steering-engine room. Combined indicators and transmitters are located on bridge deck and on quarter deck at hand-steering wheel.

Helm Angle Indicators.—Of the illuminated dial pattern, and will signal "$2\frac{1}{2}$," "5," "$7\frac{1}{2}$," "10," "15," "25" and "35" degrees each side of center "0," which has white light, with green light for starboard and red light for port. There are indicators on the bridge and quarter deck and in steering-engine room. The transmitter is on rudder head.

Electrical Revolution Indicators.—Are located on bridge and in pilot house, with transmitter on main engine shaft at after engine-room bulkhead.

Telephones.—There are telephones with push buttons, bells and annunciators, connecting the chart house to port and starboard engine rooms, Captain's cabin and steering-engine rooms; the starboard engine room to chief engineer's state room, and the executive officer's office to master-at-arms.

Ventilation.—The hull ventilation is accomplished by means of sixteen Sturtevant blowers; some of steel plate and some of cast shell type; all driven by General Electric Co.'s motors. The following is a list of sizes, service and location of all blowers.

MAIN ENGINES.

There are two four-cylinder triple-expansion engines of the vertical, inverted, direct-acting type, placed athwartship of each other in separate watertight compartments. The cylinders are arranged as shown in Fig. 1.

The cylinders are supported by twelve open-hearth nickel-steel columns, three inches in diameter, braced as shown in Fig.

A cast-steel strongback, made in two sections, for supporting the crosshead guides, runs the entire length of framing.

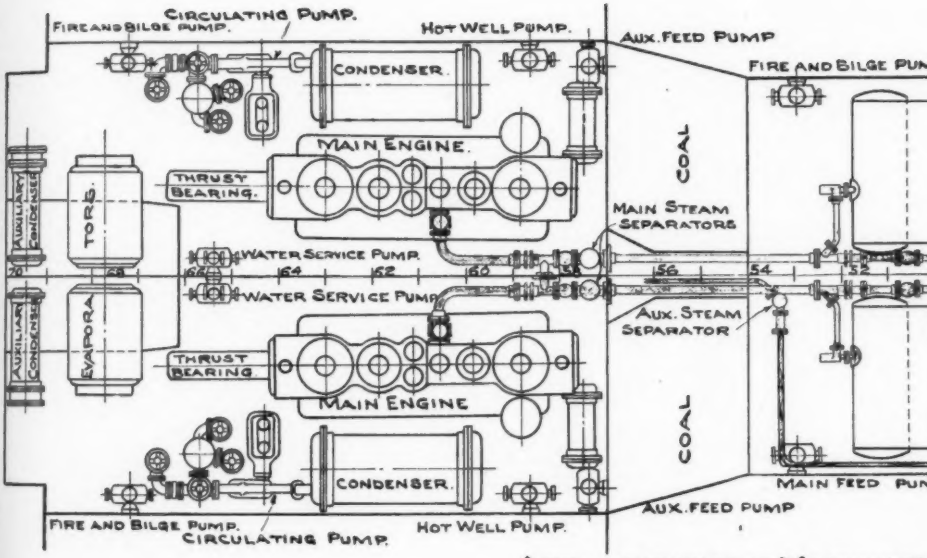
The bedplates are of cast steel, ∩ section, cast in three pieces and bolted together. The air-pump bed is cast on forward section. There are six cross girders of I section.

The H.P. valve chest of each engine is fitted with one piston valve, the I.P. valve chest with two piston valves and the L.P. valve chest with double-ported slide valves. All valves are fitted with balance pistons.

The link motion is of the Stephenson double-bar type, all valves being worked direct, with one crosshead for the I.P. valves. Adjustable cut-offs are fitted to each link with a range of from one to ten-tenths of a stroke.

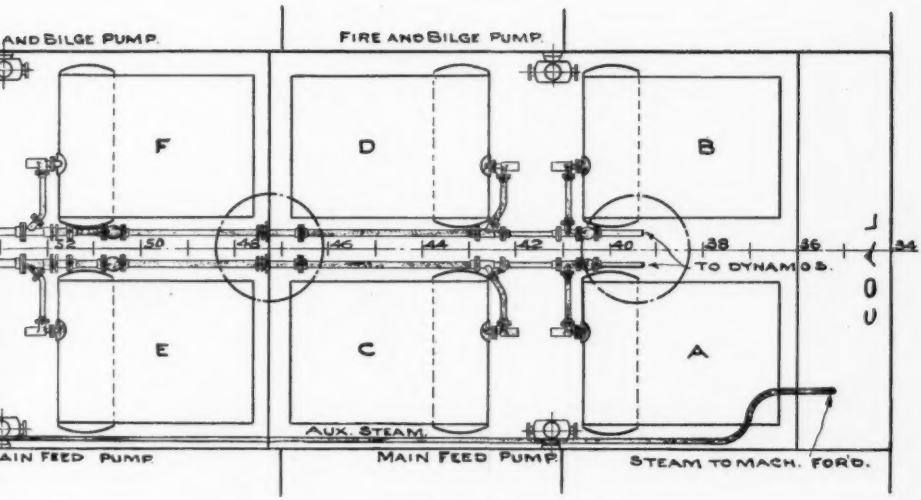
The throttle valve is of the balance-cylinder type, and operated by handwheels at the working platform.

The ahead crosshead guides are of cast iron, cored out at back for circulating water; the top end is bolted to cylinders and



ARRANGEMENT OF MACHINER
U.S. CRUISER 'CU

FIG. 1.



MACHINERY AND MAIN STEAM PIPES.
UISER "CLEVELAND"

FIG. 1.

bottom end to strongback. The backing guides are of cast iron and are bolted to the ahead guides.

The turning engine is a double, inverted, 4 × 4 × 4 engine, driving a worm shaft, which in turn drives a second worm, and which may be made at will to mesh with a worm wheel on either propeller shaft. The turning engine is made reversible by means of a change valve. A ratchet is provided for turning by hand.

SHAFTING AND BEARINGS.

The crank shafts are of forged steel, each being made in two sections, the forward section forming the forward L.P. and H.P. cranks, and the after section the I.P. and after L.P. cranks. When the sections are bolted together the cranks are 90 degrees apart, and their sequence is H.P., I.P., forward L.P. and after L.P. when in ahead gear.

The propeller shaft is fitted with a composition casing beginning 5½ inches aft the forward end of shaft and extending throughout length of shaft to 1 inch inside of propeller hub. The casings are lap jointed and made absolutely watertight. The outboard couplings are provided with composition casings filled in with melted pitch.

The thrust bearings and caps are of cast iron. There are eleven horseshoes of cast steel, lined with white metal, and adjusted by means of two 2½-inch side rods and nuts ⅝-inch thick. The end and side walls of the pedestal form a trough for the accumulation of oil. The trough is provided with a brass cooling pipe, and stuffing box to prevent the escape of oil. The pedestal is bolted to cast-iron sole plate, fitted with forged-steel wedges for adjusting fore and aft. A sheet-brass casing is fitted to the top to retain the oil.

CONDENSERS.

Each engine room is provided with a main condenser of the cylindrical type, detached from main engine and located on coal-bunker bulkhead close to protective deck. The shell is made of ½-inch boiler steel, with flanges and stiffeners of angle iron, and is 4 feet 4½ inches inside diameter and 9 feet 2 inches between

tube sheets. The tubes, tube sheets and glands are of Muntz metal, the water ends are of composition. The circulating water is on the inside of tubes. The inlet and outlet heads are provided with a valve and operating lever. There are 2,004 seamless-drawn tubes, $\frac{5}{8}$-inch outside diameter, No. 18 B.W.G., with a total cooling surface of 3,004 square feet for each condenser.

MAIN AIR PUMP.

There is a single-acting beam air pump connected to the forward L.P. crosshead of each main engine with cylinders 22 inches diameter by 10 inches stroke. The valves are made of three discs of manganese-bronze $\frac{1}{32}$-inch thick. Each pump has 18 valves.

CIRCULATING PUMP.

There is a centrifugal pump for each condenser with 12-inch inlet and outlet connection arranged to draw from the sea, bilge or main drain and discharge to the condenser or overboard through the valve in condenser head. The suction valves are fitted with a locking device, so that only one valve can be open at a time. A safety sluice valve is provided in each inboard sea pipe. Each pump is driven by a simple, vertical, two-cylinder, 6-inch by 6-inch by 6-inch engine.

AUXILIARY CONDENSER.

There is one auxiliary condenser, with combined air and circulating pump attached, in each engine room. It is of the Davidson type, with 6-inch by 7-inch by 10-innh by 10-inch pump, and 400 square feet of cooling surface.

FEED-WATER HEATER.

There is in each engine room a feed-water heater, placed horizontally, of the cylindrical type, with 353 Muntz metal tubes $\frac{5}{8}$-inch outside diameter, No. 18 B. W. G., with a heating surface of 231.04 square feet. The feed water falls by gravity through the heaters from the upper feed tanks and then to the feed-pump suction. Considerable trouble was experienced on the preliminary trials on account, at times, of cold feed water, the upper tank on one or the other side filling up and overflow-

ing into bilges. For the official trial the following arrangement was made by the contractors: The hotwell pump, instead of discharging from the lower filter tank into the upper feed tank, as designed, was fitted so as to discharge directly through the feed-water heaters into the feed-pump suction main, thus doing away with the upper tanks. A pressure gauge was fitted to the discharge of each hotwell pump, and the pressure carried at about 30 pounds by regulating the speed of the pump. This arrangement worked well but caused more or less attention, and during the official trial the feed-water heaters leaked around heads. Retarders were also fitted to tubes of feed-water heaters.

By substituting a vertical heater capable of withstanding a pressure of about 200 pounds and removing the upper feed tank a much better arrangement might be obtained and weight saved. A 2½-inch equalizing pipe connecting the upper feed tanks on each side has been recommended to be fitted, thus allowing the water to pass from one side to the other and to prevent overflowing.

DATA OF MAIN ENGINES.

Cylinders, number for each engine	4
H.P., diameter, inches	18
I.P., diameter, inches	29
F.L.P., diameter, inches	35
A.L.P., diameter, inches	35
Valves, H.P. (one for each cylinder), diameter, inches	11
I.P. (two for each cylinder), diameter, inches	11
L.P.'s slide, inches	
Valve stems, H.P. (1), diameter lower ends, inches	2
diameter through valve, inches	$1\frac{8}{16}$
diameter at balance piston, inches	$1\frac{5}{32}$
L.P. (2), diameter lower ends, inches	2
diameter through valves, inches	$1\frac{8}{16}$
diameter at balance piston, inches	$1\frac{5}{32}$
L.P.'s (1 each cylinder), diameter lower ends, inches	$2\frac{1}{2}$
diameter through valve, inches	$1\frac{8}{16}$
diameter at balance piston, inches	$1\frac{7}{16}$
Piston rods, diameter, inches	$4\frac{1}{2}$
length from piston to center crosshead pin, inches	$53\frac{1}{4}$
axial hole, inches	2
Connecting rods, center to center, inches	67
diameter upper ends, inches	$4\frac{1}{2}$
diameter lower ends, inches	$4\frac{3}{4}$
axial hole, inches	2

Main steam pipe (steel), diameter, inches	6
exhaust pipes (2) to condensers (copper), diameter, inches	12½
Crosshead pins, diameter, inches	5½
length in bearing, inches	10
Reversing gear, steam cylinder, diameter, inches	8
controlling cylinder, diameter, inches	3½
stroke, inches	12
Crank shaft, diameter, inches	9½
axial hole, diameter, inches	5
coupling discs, diameter, inches	17
thickness, inches	2½
Bolts in flanges (6), taper, mean diameter, inches	2
Crank shaft journals, diameter, inches	9½
length, inches	11
axial hole, diameter, inches	5
Crank webs, width, inches	10½
thickness, inches	5½
Thrust shaft, diameter, inches	9½
coupling discs, diameter, inches	17
thickness, inches	2½
axial hole, inches	5⅛
length, feet and inches	7– 4
collars, number, each shaft	11
diameter, inches	14
distance between collars, inches	3
thickness of collars, inches	1½
Surface, total, both engines, square inches	1,020.8
Line shaft, diameter, inches	9
axial hole, diameter, inches	5⅛
length, feet and inches	25– 0
Stern-tube shaft, diameter, inches	9
axial hole, diameter, inches	2 to 5⅛
length, feet and inches	38– 5
Propeller shaft, diameter, inches	9
axial hole, diameter, inches	2 to 5⅛
length, feet and inches	20– 3
Total length of shafting from forward end of thrust shaft to after end of propeller shaft, feet and inches	90–10½

PROPELLERS.

The propellers are designed for adjustable pitch from 10 feet 6 inches to 2 feet diameter to 11 feet 6 inches at 6 feet diameter, right-and-left, three-bladed, built up, with blades inclined aft 9 inches. The top tips of the propellers turn inboard when in ahead motion. The blades and the hubs are of manganese-bronze, tinned.

Blades, number of	3
diameter, feet and inches	10- 6
pitch as set on trial, feet and inches	11- 8
greatest width of blade, inches	43
helicoidal area of each screw, square feet	34.74
projected area of each screw, square feet	34.41
disc area of each screw, square feet	86.92
ratio diameter to mean pitch	1 to .898
distance of center of hub above lowest point of keel, feet	5.5
top of propeller immersed, feet	5.0

ENGINES 80 AND 81.

Name of vessel ..... Cruisers Nos. 14 to 19.
Total I.H.P. and displacement.....4,500 I.H.P.; 3,200 tons displacement.
Type and number of engines,
4-cylinder, vertical, inverted, triple engine, twin-screw.
Diams. of cylinders and stroke, inches..... 18, 29, 35½, 35½ by 30 stroke.
Valve gear.....Stephenson link.
Connecting rod.....67 inches between centers.
Scale of crank circle.....4 inches = 1 foot.
valve circles.....24 inches = 1 foot.

	H.P.		*I.P.*		*L.P.*	
Eccentricity, inches	2		2		2	
Travel of valve, inches	4		4		4	
Kind and number of valves and Diameter of piston valves.	One piston 11″ diam.		Two pistons 11″ diam.		One double ported slide.	
Side of valve on which steam is taken,	Inside.		Outside.		Outside.	
Width of port and length of slide valve	*Top.* 1½	*Bottom.* 1½	*Top.* 1½	*Bottom.* 1½	*Top.* 1½×37	*Bottom.* 1½×37
Steam lap	2⅝	2⅝	2⅝	2⅝	1⅝	1⅝
Exhaust lap	−5/16	−1/16	−5/16	−1/16	0	+½
Angular advance, degrees	35	35	35	35	40	40
Steam lead, angular, degrees	8	10	8	10	12	16
linear	½	5/16	½	5/16	1 1/32	1 3/32
Cut-off, in inches	23⅛	21 5/16	23⅛	21 5/16	22 1/16	20 3/16
decimal of stroke	.78	.705	.78	.705	.735	.672
Exhaust release, in inches	31 7/16	3⅛	31 7/16	3⅛	2 13/16	2⅞
decimal	.114	.1205	.114	.1205	.093	.095
Compression in inches	1⅞	1 11/16	1⅞	1 11/16	4 1/16	3⅛
decimal of stroke	.062	.064	.062	.064	.139	.129
Steam opening	1 3/32	1 5/32	1 3/32	1 5/32	1 1/16×2	1 3/16×2
Exhaust opening	Full port.		Full port.		Full port.	
Velocity of steam in feet per second at 860 feet a minute = 172 revolutions per minute	128.6	121.6	166.9	157.9	189.3	169.4
Velocity of exhaust in feet per second at 860 feet per minute = 172 revolutions per minute	93.82	93.82	121.7	121.7	134.12	134.12

Velocity of exhaust to condenser, 12½-inch nozzle, feet per second..... 115.6

BOILERS.

There are six boilers of the Babcock and Wilcox *Alert* type, in two watertight compartments. In the forward compartment, extending from frame No. 36 aft to 9 inches aft of frame No. 47, and 12 feet 5 inches either side of center of ship, there are four boilers, and in the after compartment, extending from 9 inches aft of frame 47 to frame 54, there are two boilers. A general arrangement of fire rooms is given in Fig. 2.

There are two smoke pipes for the six boilers, the forward stack taking the two forward boilers and the after stack the two middle and two after boilers.

Each boiler has a steam and water drum 42 inches inside diameter, $\frac{21}{32}$ inch thick and 10 feet 8 inches long. The tubes are set at an angle of 15 degrees to horizontal line of boiler and are expanded into vertical forged-steel headers at front and back of boiler. The upper end of each back header is connected to the water drum by a horizontal 4-inch tube, thus giving to each section an independent steam and water circulation. The headers are provided with hand holes so arranged that all tubes are easily accessible for cleaning or repairs.

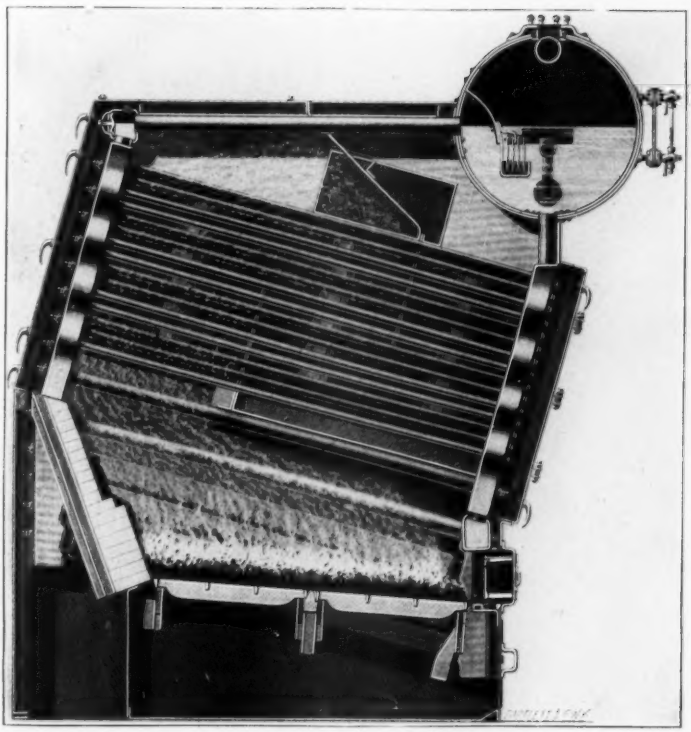
All parts of boilers under pressure are of forged steel, no cast iron or steel or malleable iron being used.

In each boiler there are 406 2-inch tubes, No. 8 B. W. G., and 32 4-inch tubes, No. 6 B. W. G. The total heating surface for each boiler is 2,200 square feet. Dusting doors are provided on the side of the boiler so that all tubes may be cleaned without hindering the steaming of the boiler.

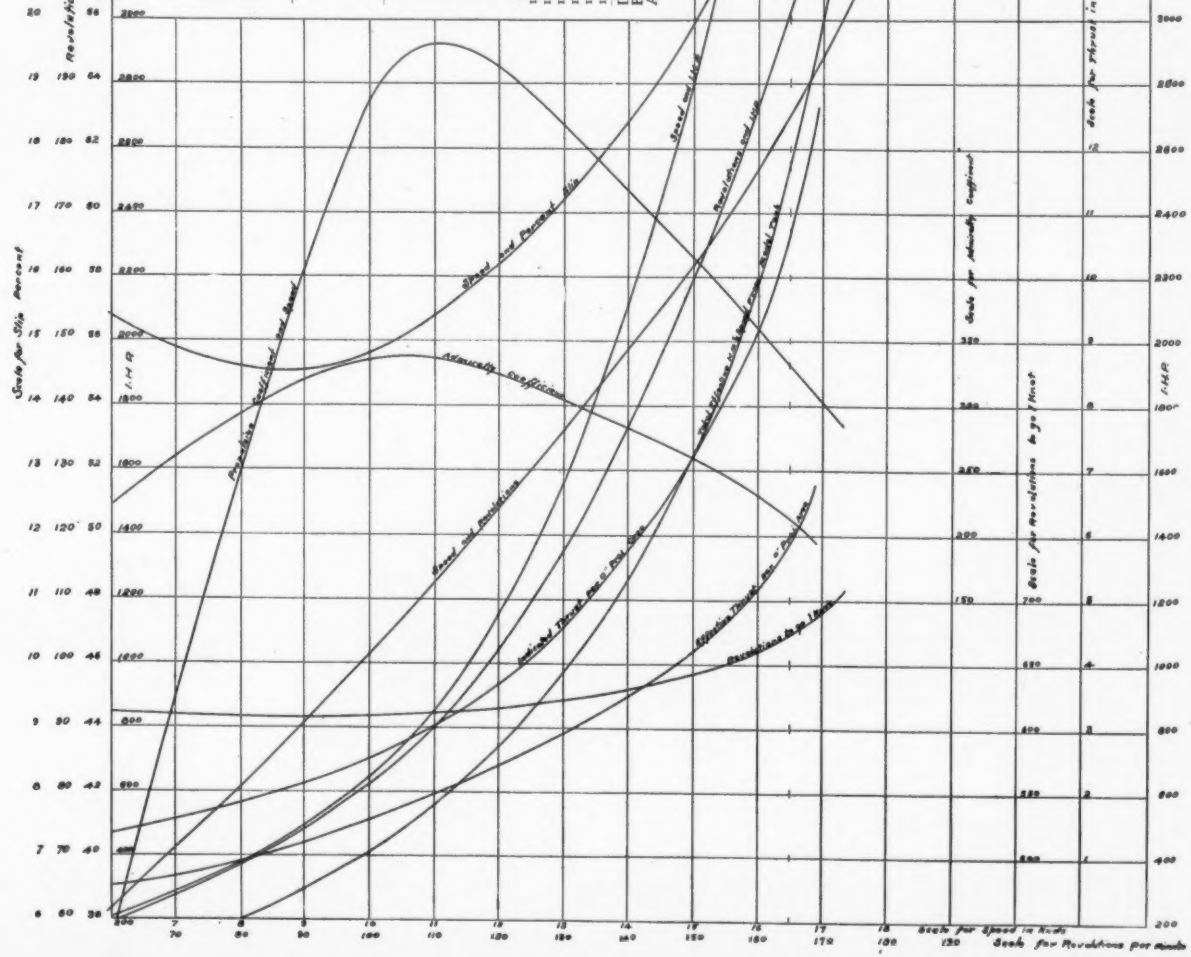
There is one rectangular furnace with three firing doors, and cast-iron grates. No fire brick is used except in bridge walls. The grate surface for each boiler is 50 square feet. The ratio of heating surface to grate surface is 44 to 1.

The stacks are 70 feet above the grates. Floor space, one boiler (about 9 feet $10\frac{1}{8}$ inches by 9 feet $\frac{1}{2}$ inch), 88.95 square feet.

Fig. 3 gives general idea of the construction of these boilers.



Babcock & Wilcox "Alert" Type Boilers.


Scale for Slip Percent
Revolutions
I.H.P.
Speed and Revolutions
Speed and Thrust
Speed and Torque
Revolutions and I.H.P.
Thrust and I.H.P.
Torque and I.H.P.
Efficiency
Revolutions to go 1 Mile
Scale for Speed in Knots
Scale for Revolutions per Minute

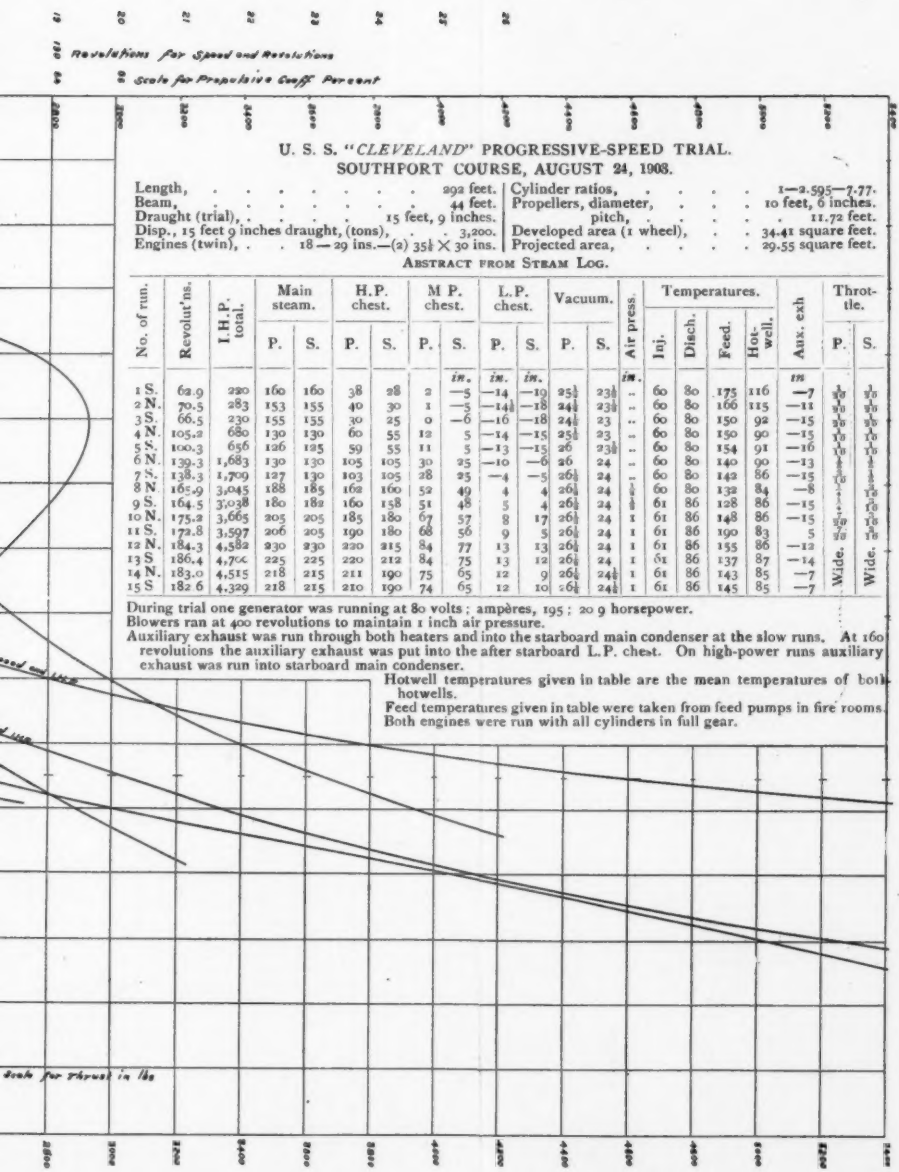
U. S. S. "*CLEVELAND*" PROGRESSIVE-SPEED TRIAL.
SOUTHPORT COURSE, AUGUST 24, 1908.

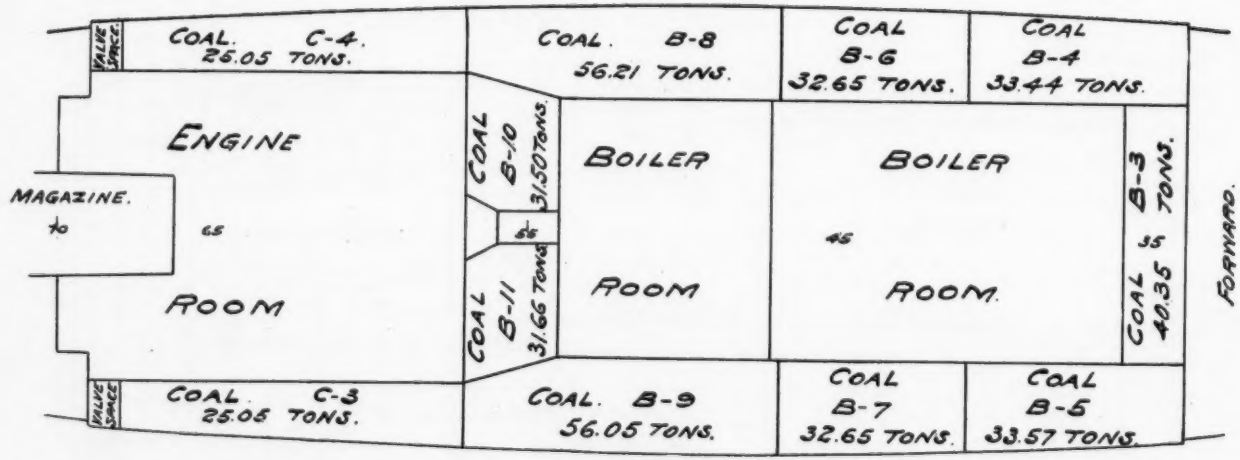
Length,	292 feet.	Cylinder ratios,	1—2.595—7.77.
Beam,	44 feet.	Propellers, diameter,	10 feet, 6 inches.
Draught (trial),	15 feet, 9 inches.	pitch,	11.72 feet.
Disp., 15 feet 9 inches draught, (tons),	3,200.	Developed area (1 wheel),	34.41 square feet.
Engines (twin),	18—29 ins.—(2) 35½ × 30 ins.	Projected area,	29.55 square feet.

ABSTRACT FROM STEAM LOG.

No. of run.	Revolutions.	I.H.P. total.	Main steam.		H.P. chest.		M.P. chest.		L.P. chest.		Vacuum.		Air press.	Temperatures.				Aux. exh	Throttle.	
			P.	S.	P.	S.	P.	S.	P.	S.	P.	S.		Inj.	Ditch.	Feed.	Hotwell.		P.	S.
							in.	*in.*	*in.*	*in.*			*in.*					*in*		
1 S.	62.9	220	160	160	38	28	2	—5	—14	—19	25½	23½	..	60	80	175	116	—7	1/10	1/10
2 N.	70.5	283	153	155	40	30	1	—5	—14½	—18	24½	23½	..	60	80	166	115	—11	1/10	1/10
3 S.	66.5	230	155	155	30	25	0	—6	—16	—18	24½	23	..	60	80	150	92	—15	1/10	1/10
4 N.	105.2	680	130	130	60	55	12	5	—14	—15	25½	23	..	60	80	150	90	—15	1/10	1/10
5 S.	100.3	656	126	125	59	55	11	5	—13	—15	26	23½	..	60	80	154	91	—16	1/10	1/10
6 N.	139.3	1,683	130	130	105	105	30	25	—10	—6	26	24	..	60	80	140	90	—13	1/10	1/10
7 S.	138.3	1,709	127	130	103	105	28	25	—4	—5	26½	24	..	60	80	142	86	—15	1/10	1/10
8 N.	165.9	3,045	188	185	162	160	52	49	4	4	26½	24	½	60	80	132	84	—8	1/10	1/10
9 S.	164.5	3,038	180	182	160	158	51	48	5	4	26½	24	½	61	86	128	86	—15	1/10	1/10
10 N.	175.2	3,665	205	205	185	180	67	57	8	17	26½	24	1	61	86	148	86	—15	1/10	1/10
11 S.	172.8	3,597	206	205	190	180	68	56	9	5	26½	24	1	61	86	190	83	5	1/10	1/10
12 N.	184.3	4,582	230	230	220	215	84	77	13	13	26½	24	1	61	86	155	86	—12	Wide.	Wide.
13 S.	186.4	4,700	225	225	220	212	84	75	13	12	26½	24	1	61	86	137	87	—14		
14 N.	183.0	4,515	218	215	211	190	75	65	12	9	26½	24½	1	61	86	143	85	—7		
15 S.	182.6	4,329	218	215	210	190	74	65	12	10	26½	24½	1	61	86	145	85	—7		

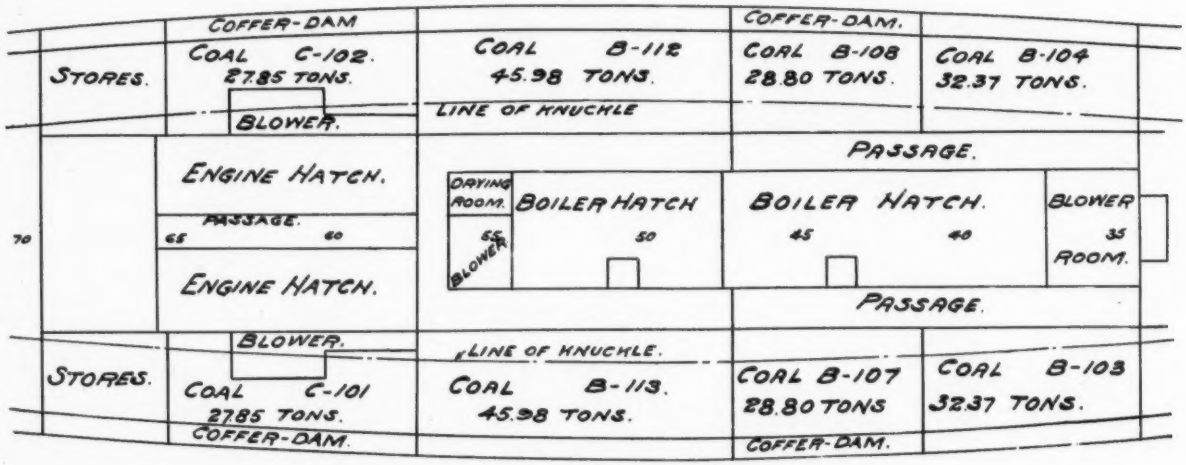
During trial one generator was running at 80 volts; amperes, 195; 20.9 horsepower.
Blowers ran at 400 revolutions to maintain 1 inch air pressure.
Auxiliary exhaust was run through both heaters and into the starboard main condenser at the slow runs. At 160 revolutions the auxiliary exhaust was put into the after starboard L.P. chest. On high-power runs auxiliary exhaust was run into starboard main condenser.
Hotwell temperatures given in table are the mean temperatures of both hotwells.
Feed temperatures given in table were taken from feed pumps in fire rooms.
Both engines were run with all cylinders in full gear.





COAL STOWAGE IN HOLD.

FIG. 3.



COAL STOWAGE ON PROTECTIVE DECK

FIG. 4.

VENTILATING SETS FOR THE U. S. CRUISER *CLEVELAND.*

No. of set.	No. of fan.	Classification.				Cubic feet per min. at 1½ ounces.	Supply or exhaust.	Weight of set.	Weight of motor.	Location.		
		Mark.	Pole.	H.P.	R.P.M.					Deck.	Frame.	Side.
1	1	C-A	2	¼	2,230	342	E	235	135	Gun.	11	Port.
2	5	C-E	2	1½	1,220	1,210	E	*725	335	Berth.	16	"
3	6	C-E	4	2	1,030	2,150	E	*1,000	465	"	18	Starb.
4	4	C-A	2	¾	1,650	967	E	*475	240	"	27	Centr.
5	3	C-A	2	½	1,650	695	S	*325	138	Prot.	29	"
6	6	C-E	4	2	1,030	2,150	S	*1,000	465	"	30	Port.
7	6	C-E	4	2	1,030	2,150	S	*1,000	465	"	30	Starb.
8	4	C-A	2	1	1,390	967	E	*475	240	"	32	Centr.
9	50″	M-P	4	4	900	4,520	S	1,160	548	Engine Room		Starb.
10	50″	M-P	4	4	900	4,520	S	1,160	548	"	"	Port.
11	3	C-A	2	½	1,650	695	E	*325	138	Berth.	65	Starb.
12	5	C-E	2	1½	1,220	1,210	S	725	335	Prot.	70	Centr.
13	4	C-A	2	1	1,390	967	S	*475	240	"	81	"
14	3	C-A	2	½	1,650	695	S	*325	138	"	87	Port.
15	3	C-A	2	½	1,650	695	E	*325	138	Berth.	97	Centr.
16	6	C-E	4	2	1,030	2,150	E	*1,000	465	"	95	Port.

NOTE.—Number of set is the number of blower in ship.
Number of fan is the Sturtevant Monogram.
Cast-iron shell.
The 50″ fans are steel-plate type.
Classified by General Electric Co.
Weights marked thus * are approximate, others are actual.
Deliveries are with restricted outlet of $\frac{DW}{3}$.

AUXILIARIES.

Forced Draft.—The forced draft is of the closed fire-room type, air being furnished by three 5-inch by 5-inch by 5-inch, two-cylinder, simple, inverted, direct-acting blower engines with 60-inch fans. The engines and fans are of the contractors' design. The fans have induction nozzles with an area of 1,134.12 square inches, and discharge from the periphery of the fan direct into the fire room.

Main Feed Pumps.—There is in each fire room one 10-inch by 7-foot by 12-inch vertical double-acting plunger pump of the Davidson type. Each main feed pump is piped to draw from the feed tanks, the hotwell tanks and the reserve feed tanks and to discharge into the boilers or reserve feed tanks. The aux-

iliary feed pumps are of the same size and pattern as the main feed pumps. There is one on the forward bulkhead in each engine room piped to draw from feed tanks, hotwell tanks, the sea, the drainage system and the engine-room bilge, and will discharge into boilers, fire main, overboard or to ash ejectors in fire rooms.

Fire and Bilge Pumps.—There is in each engine room one 7-inch by 7-inch by 12-inch single-cylinder vertical pump, Davidson type, piped to draw from the sea, engine-room bilge and the drainage system and discharge to fire main or overboard.

Water-Service Pump.—There is in each engine room one 7-inch by 7-inch by 12-inch single-cylinder vertical pump, Davidson type, piped to draw from the sea and discharge to engine-room water service, fire-main or distiller circulating pipe, through which the water may be passed through the distiller or around it to the flushing main or overboard.

Hotwell Pump.—There is in each engine room one 7-inch by 7-inch by 12-inch single-cylinder vertical pump, Davidson type, piped to draw from hotwell, air-pump suction pipe, the reserve feed tanks and from sides of ship above water line for taking fresh water for boilers, and to discharge into feed tanks and reserve feed tanks.

Fire-Room Bilge Pump.—There is in each fire room one 7-inch by 7-inch by 12-inch single-cylinder vertical pump, of the Davidson type, piped to draw from the sea, from the bilge of their own compartment, and from the boilers, and discharge into the sea, the fire main, to the reserve-feed tanks and boiler-tube cleaning piping.

Shaft-bilge Pump.—There is a bilge pump in each engine room, driven from a crank pin on the forward end of the main engine shaft, with a water cylinder 5 inches in diameter and 5¾ inches stroke, and draws from crank pits and bilges and discharges overboard.

Hotwell and Filter Tank.—There is a hotwell and filter tank of 1,150 gallons capacity under condensers in each engine room. It is provided with discharge connections from the main and auxiliary air pumps, hotwell pump and drain traps, and suction connections for feed pumps and hotwell pumps.

Feed Tanks.—Each engine room is provided with a feed tank

of 500 gallons capacity, located in forward end of engine-room casing, between gun and protective decks. Each tank is provided with discharge connections from main and auxiliary air pumps and hotwell pump, and an outlet to the feed-water heater and a vapor pipe. The feed tanks have floats and levers for regulating the hotwell pump throttle.

Engineers' Workshop.—The workshop is located on berth deck forward between frames No. 25 and No. 31½, and contains one 24-inch lathe, one 11-inch lathe, one 21-inch upright drill, one 18-inch shaper, two 12-inch emery wheels and one grindstone, all driven by a 4-inch by 4½-inch upright Sturtevant engine.

Ash Hoists.—There are two double-cylinder 4½-inch by 4½-inch by 4½-inch Williamson's reversing ash hoists, one in each boiler casing between main and gun decks, for hoisting ashes through the ventilators. They will raise a bucket of ashes weighing 300 pounds in five seconds, with 80 pounds of steam.

Distilling Plant.—The distilling plant, located in after ends of engine rooms, consists of two evaporators and two distillers, one evaporator feed pump and two distiller circulating pumps, all of Davidson pattern. The evaporators are so arranged that they may be worked with double effect, and each has a capacity of 5,000 gallons per 24 hours. The evaporator feed pump is a vertical, 4½-inch by 4-inch by 6-inch pump, and draws from the overflow of the distiller circulating water and from the sea, and is piped to feed both evaporators. The distiller circulating water is supplied by the engine-room water-service pump. The fresh-water and brine pump is a combined 4-inch by 2⅜-inch by 2⅜-inch by 5-inch horizontal pump.

Steering Engine.—The steering engine is a double-cylinder, 8-inch by 8-inch by 10-inch engine of the Hyde Windlass Co.'s design. It is located in a separate compartment on platform deck, aft, and is connected by screw gears to the rudder post.

Windlass.—There is a double-cylinder, 9-inch by 9-inch by 9-inch windlass of the Hyde Windlass Co.'s design, located on the gun deck, forward, at frame No. 15.

Electric Winch.—Two 25-H.P. electric winches of the General Electric Co.'s series-wound type, are located on main deck, one forward at frame No. 32 and one aft at frame No. 74.

Air Compressor.—The air compressor for operating the fourteen watertight doors of the long-arm system is located in blower room on platform deck forward. It is of the vertical, triplex, single-acting, three-stage type, with cylinders $1\frac{3}{8}$ inches by $2\frac{5}{8}$ inches by $5\frac{1}{2}$ inches diameter, with 4-inch stroke, and is driven by a $7\frac{1}{2}$-inch H.P. electric motor. The compressor is fitted with an inner cooler, and on same bed with motor.

Heating System.—The heating system is composed of five steam circuits. Circuit No. 1 supplies steam to gun deck forward; No. 2 to gun deck midship; No. 3 to gun deck aft; No. 4 to berth deck forward, and No. 5 to berth deck aft. Pilot house and chart room are supplied from circuit No. 1.

Circuits Nos. 2, 3 and 5 are taken from the auxiliary steam in engine room, and circuits Nos. 1 and 4 from steam to dynamos. The galley circuit is taken from the auxiliary steam in forward fire room. The pantry circuit is taken from the auxiliary steam in engine room, and has a branch to the economic heaters in Captain's, ward room, and sick-bay bath room.

Brass-pipe coil radiators are used throughout the ship.

The following is a tabulated list of weights of certain parts of the U. S. S. *Cleveland:*

	Tons.
1. Machinery, auxiliaries and piping in engine rooms	148.49
2. Propellers and shafting	25.61
3. Boilers, smoke pipe and uptakes, No. 1 fire room	110.
4. Boilers, smoke pipe and uptakes, No. 2 fire room	53.40
5. Evaporator plant	8.01
6. Refrigerating plant	4.57
7. Heating system	8.55
8. Anchor engine	8.48
9. Anchor engine piping	.62
10. Steering engine	5.42
11. Steering engine gear	1.15
12. Steering engine piping	.73
13. Generators	13.31
14. Generators' piping	1.45
15. Ventilating plant	16.27
16. Ventilating plant for bunkers	3.37
17. Ventilating blowers	2.35
18. Ventilating motors	1.83
19. Long-arm system water-tight doors	6.05

ELECTRIC PLANT.

The electric-light and power equipment is entirely furnished by the General Electric Co., and consists of generators, incandescent lights and power apparatus.

There are four generating sets, each of 24 kilowatts capacity, compound wound, running at 80 volts, 300 ampères. Each of these sets is self-contained and is driven by a cross-compound engine with cylinder diameters 6½ inch and 10½-inch by 7-inch stroke, running at 400 revolutions with 100 pounds steam. Each of these generating sets is connected to the main switchboard in the dynamo room, which is so arranged that any dynamo can distribute current individually to the searchlights, power or lighting circuits, or they may work on any or all of the circuits in conjunction with each other. Three hundred and forty-five outlets are provided on the lighting circuit, 16 of the outlets being for fans in the state rooms, the lighting circuits being fed by 8 feeders. One Ardois night signal set is fitted to the former, with a keyboard on the bridge. Truck lights are installed on each mast. Cargo reflectors, diving lamps and running lights are also furnished. Two 30-inch, hand-controlled searchlights are mounted upon the platform above the bridge. The lamps are of the horizontal type with combination automatic and hand feed, and are designed for a current of from 70 to 80 ampères, with a difference potential across the arc of about 50 volts.

These searchlights are fed by independent feeders through circuit breakers and rheostats at the main switchboard and have separate busbars.

AUXILIARY MACHINERY AND INDICATED HORSEPOWER OF MOTORS.

		I.H.P.	Total H.P.
1	Long-arm compressor motor,	2½	2½
10	Chain ammunition hoists,	2½	25.
16	Ventilating fans, varying,	30.	30.
2	Deck winches, rated at 25 H.P.,		50.
	Total,		107.5

LONG-ARM DOORS.

This system is complicated by a great number of joints in the piping, causing leakages, and is unnecessarily heavy.

The joints between the doors and bulkheads are not tight, and the contractors have been unable to make them so.

The door framing should be heavy enough to make the doors watertight and the door should be tested with water jacket and not by gauges. This plant is very liable to get out of order, and will necessitate considerable attention at all times.

The system should be done away with and an electric plant substituted.

The following tests of the long-arm system were made:

The doc , were tested with a line pressure of 150 pounds per square inch, and the time required to close and lock varied from 6 to 30 seconds. The time to open and lock varied from 11 to 45 seconds.

The test of the vertical doors by the hand pump showed the time to close and lock varied from 39 seconds to 120 seconds and to open and lock from 68 to 150 seconds.

The horizontal doors by hand pump showed time to close and lock varied from 13 to 40 seconds and to open and lock from 10 to 40 seconds. There are four of the fourteen doors which can be operated from one side only.

The fit of the doors in coal bunkers was then tested by a feeler gauge, and .004 inch was taken as the limit. In some cases .012-inch gauge entered, and in other cases .004-inch would not enter. The gauges were pressed between the doors and the frames at eight different points.

The piping was also tested for tightness.

The receiver was pumped up to 950 pounds per square inch and in two hours the pressure in the receiver had fallen to 709 pounds per square inch. The total time to charge the receiver was about two hours.

CURVES OF I.HP., REVOLUTIONS, SPEED, ETC.

On August 24, 1903, a series of fifteen progressive runs were made over the measured-mile course off Southport, Me. The course had been laid down for the progression trials of the torpedo boats *Bagley*, *Barney* and *Biddle*, built by the Bath Iron Works. The marks on shore were not very distinct, but the

runs and data obtained were taken under the direction of the contractors, whose request for an official standardizing of the propellers had not been granted. Previous to these runs 525 tons of coal had been stowed on board, at the Government coaling station at East Lamoine, Me., and the bunkers sealed, in order to bring the vessel to about the trial displacement of 2,300 tons.

With the exception of a preliminary run to Boston and return, for docking and to East Lamoine, Maine, no full-speed trials were made. These trials were run with the vessel light, and only on the return from Boston, after docking, and during the standardization trials, were all the boilers in use. The vessel went on her official trial without much preliminary training of the fire-room force, and this may account for the variation of the steam pressure during the official trial.

On the completion of the runs the vessel was headed for Bath, Maine, and it was intended to have a full-speed run of about two-hours' duration, but the starboard engine began to knock badly around the H.P. cylinder and the engine was stopped. The remainder of the run to Bath was continued with the port engine in use. On removing the H.P. cylinder head and piston a $\frac{3}{4}$-inch bolt $1\frac{3}{4}$ inches long was found on the bottom of the cylinder hammered down to $\frac{3}{8}$ inch on head. The piston ring was found jammed on one side and the flanges and followers of piston eased off. The bottom of the cylinder and the piston were indented, but there was no further damage to any part. It is a question how the bolt got into the cylinder without doing more damage, and it is surmised that it laid in the cylinder port and thus worked into the cylinder without passing the valve.

If the speed and revolution curve is correct the vessel should have made the required speed, judging from the average revolutions made on the official trial. Presuming that the buoys are accurately placed, the vessel failed by .0494 knots on the four-hours' run.

The method of standardization is liable to error and allows more or less jockeying, and I consider a trial over a measured course for a number of hours less liable to error. In case of

OFFICIAL TRIAL, U. S. S.

No. of run.	Time.	Time interval.	Reading of counter.	Revolutions during interval.	Revolutions per minute.	Steam pressures. In boilers above atmosphere.	In st'm pipe at engine above atmosphere.	In H.P. steam chest, absolute.	In 1st I.P. receiver, absolute.	In L.P. receiver, absolute.
	h. m. s.	*m. s.*								**PORT**
First.	12'30'10	...	257,572	...	...	266	240	251.7	104.7	28.7
	12'45'10	15	260,393	2,821	188	246	230	261.7	106.7	29.7
	1'00'10	15	263,154	2,761	184	253	224	228.7	94.7	25.7
	1'15'10	15	265,944	2,790	186	244	231	234.7	94.7	28.7
	1'30'10	15	268,682	2,738	182.5	243	220	225.7	94.7	26.7
	1'45'10	15	271,450	2,768	184.5	258	235	239.7	94.7	27.7
	2'00'10	15	274,208	2,758	183.8	251	235	241.7	94.7	29.7
	2'15'10	15	276,980	2,772	184.8	243	224	229.7	94.7	28.7
	2'30'00	14'50	279,725	2,745	185	...	225	229.7	94.7	28.7
Second.	2'43'37	...	282,460	...	...	248	220	229.7	94.7	28.7
	2'58'37	15	285,215	2,755	183.6	240	225	229.7	94.7	28.7
	3'13'37	15	287,976	2,761	184	232	212	214.7	89.7	26.7
	3'28'37	15'30	290,720	2,744	182.9	248	212	218.7	89.7	26.7
	3'43'37	15'30	...	...	...	261	...	...	...	...
	3'58'37	15'30	296,162	5,442	181.4	256	235	239.7	94.7	29.7
	4'13'37	15	298,945	2,783	185.5	262	235	239.7	94.7	29.7
	4'28'37	15	301,780	2,835	189	264	230	234.7	96.7	28.7
	4'43'37	15	304,615	2,835	189	264	231	234.7	96.7	28.7
	4'45'00	1'23	305,050	...	...	...	...	...	...	...
										STARBOARD
First.	12'30'10	...	441,840	...	...	...	225	224.7	94.7	26.7
	12'45'10	15	444,680	2,840	189.3	266	225	224.7	94.7	26.7
	1'00'10	15	447,470	2,790	186	246	220	224.7	89.7	25.7
	1'15'10	15	450,250	2,780	185.3	253	240	234.7	94.7	28.7
	1'30'10	15	453,040	2,790	186	244	215	214.7	87.7	24.7
	1'45'10	15	455,790	2,750	183.3	243	235	224.7	95.7	26.7
	2'00'10	15	458,601	2,811	187.4	258	245	239.7	99.7	27.7
	2'15'10	15	461,382	2,781	185.4	257	230	229.7	92.7	26.7
	2'30'00	14'50	464,149	2,767	186.5	243	235	234.7	94.7	26.7
Second.	2'43'37	...	466,692	...	...	...	235	224.7	89.7	25.7
	2'58'37	15	469,500	2,808	187.2	248	240	229.7	92.7	26.7
	3'13'37	15	472,220	2,720	181.3	240	230	224.7	89.7	25.7
	3'28'37	15	474,970	2,750	183.3	232	215	219.7	82.7	24.7
	3'43'37	15	477,775	2,805	187	248	240	239.7	94.7	26.7
	3'58'37	15	480,570	2,795	186.3	261	245	244.7	96.7	26.7
	4'13'37	15	483,380	2,810	187.3	256	245	244.7	96.7	26.7
	4'28'37	15	486,205	2,825	188.3	262	240	239.7	94.7	26.7
	4'43'37	15	489,092	2,887	192.5	264	245	243.7	97.7	28.7
	4'45'00	1'23	489,357	2,265	191.5	...	245	244.7	95.7	27.7

Coal used on trial, 37,400 pounds. Coal per

NOTE.—Port engine.—At 3'30 the vacuum dropped to 7 inches. Electric lights went out. Air pump hammered badly. Engines were slowed down to almost a stop. The gate valve in air pump suction was found shut. At 3'52 engines were opened out to full speed.

CLEVELAND, SEPTEMBER 1, 1903.

Vacuum.	Link.	Temperatures.						Revs. or double strokes per minute.						Air pressure in inches of water.	Bilge pump.
		E.R. work'g platform.	F.R. working level.	Injection.	Discharge.	Hotwell.	Feed.	Air pump.	Circulating pump.	Sanitary pump.	Feed pump.	Engine room blowers.	Fire room blowers.		
ENGINE															
27		...	...	...	...	...	...		...				...	...	...
26¾	Full gear.	93	...	64	96	90	160	Attached.	350	...	28	Electric.	395	1½	60
26¾		...	...	...	...	...	...		...	...	...		...	...	...
26¾		...	...	64	98	89	155		250	...	28		395	1½	...
27		97	...	...	...	...	...		...	...	...		...	...	...
26¾		...	...	64	98	90	150		260	...	30		395	1¼	...
26¾		...	...	...	...	...	...		...	...	...		...	...	...
27		96	...	64	97	89	140		320	...	27		395	1	...
26¾		...	...	...	...	...	...		...	...	...		...	...	...
27		...	...	...	...	...	...		...	...	...		...	...	...
27	Full gear.	95	...	64	97	90	190	Attached.	300	...	24	Electric.	400	1	...
26¾		...	...	...	...	...	...		...	...	...		...	...	...
26¾		...	...	64	98	90	162		250	...	27		400	1½	...
...		96	...	...	...	...	...		...	...	...		...	...	...
27		...	...	64	97	89	178		240	...	32		400	1 1/16	...
27		...	...	...	...	...	...		...	...	...		...	...	...
27		98	...	64	98	90	128		245	...	30		400	1½	...
27		...	...	...	...	...	...		...	...	...		...	...	...
...		...	...	...	...	...	...		...	...	...		...	...	...
ENGINE															
25		...	...	...	...	...	...		...	...	...		...	...	...
25		92	...	64	98	94	160		162	68	28		390	1¾	62
25		...	...	64	96	94	155		160	65	28		...	1¾	62
25		...	98	64	97	96	150		155	62	28		400	1½	62
25		98	...	64	100	94	140		150	60	30		...	1 1/16	63
25		...	93	64	98	94	190		149	60	24		390	1¾	63
25		...	...	64	98	94	162		152	60	27		...	¾	61
25		96	93	64	101	94	178		156	62	28		390	...	60
25		...	...	64	102	96	128		155	64	28		...	1	60
25		96	...	64	102	94	140		148	62	...		...	...	60
25		...	...	64	100	94	150		150	62	24		390	1	58
25		...	...	64	98	98	140		148	62	27		...	1	56
25		...	...	64	100	96	135		158	62	30		...	1½	58
25		104	...	64	102	96	130		150	62	32		390	1	60
25		...	...	64	101	96	130		148	61	32		...	1	60
25		...	...	64	100	85	122		150	60	32		390	1 1/16	58
25		...	100	64	106	76	128		152	60	31		...	1 1/16	59
25		90	...	64	103	86	120		154	62	30		400	1⅛	58
25		...	...	64	98	90	125		150	62	...		...	...	58

hour, 9,350 pounds. Kind of coal, Pocahontas.

Starboard engine.—Auxiliary exhaust into L.P. valve chest. Hotwell pump running at various speeds, pumping the feed water through the heater to main feed-pump suction, keeping a pressure of 10 to 30 pounds. Steam on H.P. jacket. Throttle wide open. No live steam to receivers.

interference of other vessels on the course there should be no objections to one or more vessels of the torpedo-boat class patrolling the course.

OFFICIAL TRIAL.

At 9·45 A. M., September 1st, 1903, the vessel got under way from Portland Harbor and proceeded under easy steam, with all boilers, toward the northern end of the Cape Ann trial course.

The sea was smooth with a long swell from south and east, slack water and a light wind from southeast. The conditions were very good for a trial. About 15 minutes before the first stake boat was reached the fire rooms were closed and the revolutions gradually increased.

At 12·37·20 P. M. the first stake boat was passed and the trial began. Shortly after passing this stake boat one of the safety valves began to blow and the boiler furnace doors were opened for an interval of about 5 or 10 minutes. The end of the first leg of 33 nautical miles was reached at 2·37·13.5, without any mishap, and everything worked well although the steam pressure had been somewhat variable. The vessel was sheered with a starboard helm for the turn, and the helm was put to port 5 degrees. At 2·50·56 the stake boat was passed on the return trip.

Between the 3d and 4th buoys a three-masted schooner crossed the track, and the vessel sheered off the course in order to clear the schooner. This incident caused more or less of a delay, and some allowance was made by the Board of Inspection. The vessel was also off the course for a short time, due to a light fog, when one of the stake boats could not be seen.

At about 3·30 P. M. the vacuum of the port engine dropped to 7 inches and the engine slowed.

It was found that the gate valve in the suction of the port air pump had partly closed. The valve was opened and the air pump began pounding badly, due to an overload and to opening the valve too quickly. The vacuum improved, and in about 20 minutes the engines were working together. The air pump continued pounding during the remainder of the trial. The crosshead guide of the air pump heated and was cooled off with water.

At 4·52·11 P. M. the last stake boat was reached and the trial completed.

A series of turning trials and tests of the anchor engine were made, the tactical diameter being estimated to be about 350 yards.

The vessel proceeded to Portland Harbor for anchorage.

The mean displacement was about 3,202 tons.

The port I.P. and starboard H.P. crosshead brasses ran very warm. Water was used on the L.P. ahead eccentrics and thrust bearings only.

With the above exceptions the machinery worked very well, and the boilers with perhaps more regularity and satisfactory firing would have furnished sufficient steam to have maintained the contract speed. The following auxiliaries were in use:

2 Main feed pumps.

1 Auxiliary feed pump (to be used in the feed system in case of an emergency).

2 Main circulating pumps.

2 Fire and bilge pumps.

1 Water-service pump.

2 Hotwell pumps.

3 Forced-draft blower engines.

1 Dynamo and steering engine.

The auxiliary exhaust was open to the starboard second receiver. All stuffing boxes of the main engines were fitted with metallic packing.

PERFORMANCE.

Steam pressure at boilers (per gauge), pounds	251.49
H.P. steam chest, S. engine (per gauge), pounds,	229
P. engine (per gauge), pounds,	227
1st receiver, starboard engine, absolute, pounds..	91.79
port engine, absolute, pounds	95.64
2d receiver, starboard engine, absolute, pounds...	28.3
port engine, absolute, pounds	26.6
Vacuum in condensers, in inches of mercury, port	26.9
starboard	25.
Mean effective pressures in cylinders, in pounds per square inch:	
Main engines, starboard, H.P.	102.06
1st I.P.	44.57
forward L.P.	13.025

Mean effective pressures in cylinders, in pounds per square inch :	
Main engines, starboard, aft L.P.	14.24
port, H.P.	92.9
1st L.P.	46.75
forward L.P.	14.05
aft L.P.	14.125
Mean pressure, in pounds per square inch on L.P. piston, equivalent to aggregate M.E.P. on all pistons, starboard.	42.23
port	42.405

Auxiliary Machinery.

Pump, air, main	Attached.
auxiliary	Not used.
circulating, starboard, main cylinders, outboard	27.50
inboard	28.875
port, main cylinders, outboard	39.56
inboard	41.25
Revolutions per minute :	
Main engines, starboard	186.4435
port	185.346
Pumps, air, starboard	Attached.
port	Attached.
circulating, starboard	152.6
port	276.87
feed, double strokes per minute, starboard	28.68
port	28.25
auxiliary, sanitary	62
bilge	59.88
blower engines, fire room	395
Speed of ship in knots per hour, corrected	16.4506
Slip of propeller in per cent. of its own speed, based on mean pitch :	
Starboard	23.69
Port	23.34
Air pressure in fire rooms	1.03
ash pits	1 inch of water.

Indicated Horsepower.

	Starboard.	*Port.*
Main engines, H.P.	713.8	646.54
1st I.P.	822.45	858.11
F.L.P.	361.58	386.73
L.P.	395.37	401.82
total	2,293.20	2,293.20
Air pump, starboard engine	Attached.	
port engine	Attached.	
Circulating pump, starboard engine	7.237	
port engine	19.528	

Bilge pumps, starboard	.7197
port	.7197
total	1.4394
Sanitary pumps, starboard	.7941
port	.7941
total	1.5882
Feed pumps, two main pumps, total	27.03
Ventilating engines for ship and for engine rooms	Electric.
Blower engines for forced draft in fire rooms, 3 blowers, H.P. each	6.22
total	18.66
Dynamo engines, No. 1 in use during trial, horsepower	22.00
Collective of all main engines	4,586.40
main engines, air, circulating and feed pumps	4,640.19
and auxiliary engines in operation during trial	4,684.8826
all machinery during trial, per square foot of G.S.	15.616
H.S.	.3584
Main engines, air, circulating and feed pumps, per sq. ft. of G.S.	15.4673
H.S.	.3515

COAL.

Kind and quality	Pocahontas, hand-picked.
Pounds per hour	9,350
per I.H.P. collective, all machinery in operation	1.995
main engines, air, circulating and feed pumps	2.015
square foot of G.S.	31.169
H.S.	.708
Cooling surface, square foot per I.H.P.	1.28
Heating surface, square foot per I.H.P.	2.817

Previous to the trial a Klinger reflex water gauge was fitted to each boiler in place of the mica water gauges and gave satisfaction. The Bureau of Steam Engineering has since approved the fitting of these gauges in place of the mica gauges usually called for by the machinery specifications.

After the trial the port air pump was opened up and was found that the air-pump bucket was cracked and bent, the pump rod and crosshead bent, the nut on the bucket end of rod loose, and the bucket valves more or less damaged. A new bucket and valves were fitted, the pump rod and crosshead were straightened.

Upon examination the boiler tubes and drums were found in excellent condition. A number of the front and back header plates had been leaking more or less, and this is one of the great defects of this type of boiler; for while the plates may be tight

while under steam, there is a chance of losing considerable water from leakage, and the consequent wearing away of the material of the header sheet when not under steam.

The main engine cylinders and parts were found in good condition.

The port after low-pressure crank pin and brasses were scored.

The starboard H.P. and port I.P. crosshead pins and brasses were also scored.

Considerable water leaked around the packing of the main feed pumps and so was lost in the bilges. This is the chief objection to the outside-packed plunger pumps. Either the stuffing boxes of the water ends should be made deeper or else the ordinary piston-packed pumps should be substituted.

The composition liner of one of the water-service pumps was badly scored and had to be renewed.

The fitting of composition or Tobin-bronze piston rods to the steam ends of auxiliary pumps should be omitted; in nearly every case these rods were found scored.

The average revolutions of both engines by the deck counter on the south run were 186.41 per minute, and the average on the north run 185.49. The ship practically made the speed on the south run and fell behind on the north. The conditions as regards wind and weather being more favorable on the north run it is surmised that the slowing of the port engine made the difference in speed.

NOTES AND COMMENTS.

During the trial there was a great need shown for rather extensive splash plates, of light galvanized iron, around the working parts of the engines. This, of course, would cover, to a great extent, the moving parts, but with the canvas screens, as required by the Bureau of Steam Engineering, there was absolutely no cleanliness around the working platforms.

It will be noted that these engines turn inboard and that the starting platform is inboard along the center line.

There appears to be a slight reduction in the maneuvering power due to inboard-turning propellers, in fact, the effect, as far

as could be judged, seemed to be that the ship was a little slow in turning when lying still in the water.

There is no reason why the propellers should not turn outboard and still retain the center-line working platform so that there would be practical control of all the engines by the bulkhead door between the engine rooms.

There appears ample room for everything on board with the exception of the main engines and their auxiliaries, and the engine room spaces are very much cramped.

The fitting of automatic devices to the boiler stop valves, if necessary, should be as simple as possible. Steam gauges fitted on both sides of the valve near operating valves are required in order to properly test out, and if the device is fitted it should be kept in working order and not disconnected. The arrangement should be thoroughly tested in shop before attaching to the boilers, as well as tested when fitted up on board ship.

The arrangement on this vessel I consider very complex and liable to derangement.

The lead of the auxiliary steam piping is complicated, and would have been better arranged had there been a connection direct from the main steam piping in after fire room. There is in each engine room a main steam-pipe separator (see plan of steam piping) with a nozzle for the auxiliary steam which passes around the engine room and also to the fire rooms through a second separator in the after fire room. This second separator appears to me to be unnecessary.

A valve or valves is necessary in branch main steam pipe connecting the two engine rooms in order to keep steam off one or the other engines if one is needing repairs while under way or when in port. As now arranged the throttle valve is all that shuts the steam off either engine when the auxiliary system is in use, and a throttle valve is not always tight. This change has been recommended to be made at a Navy Yard and approved by the Bureau of Steam Engineering.

The height of the boiler dead plates of furnace door above the fire room floors vary from 19 inches to 22 inches. This is not sufficient for quick firing, and should be at least 26 inches.

As now fitted I would recommend doing away with the locking device attached to the main-engine, sea and bilge injection valves. In either case the valves can be opened without making use of the key of the locking device. By using a file instead of the key, or by exerting sufficient power and straining and distorting the gland of the stuffing box and the attachment in general the valves can be opened. This latter has happened twice on board the *Cleveland*, and the gear had to be taken to the shop to be overhauled. It causes unnecessary complication, and in case of an emergency the key may have been mislaid.

The bunker ventilating pipes are excessively heavy, being composed of two 4½-inch galvanized iron pipes in the engine rooms and two in the fire-room hatches. In the engine rooms these supply pipes extend from the upper part of the hatches down to the protective deck, with 3-inch branches to the coal bunkers adjacent. Practically there is the same arrangement in the fire rooms. The exhaust pipes from the bunkers are made of a lighter weight of material above the protective deck, and of 4-inch galvanized iron piping up to the protective deck from the coal bunkers.

The main circulating pumps were tested on board ship with the engine and fire-rooms compartments partly flooded. The water was pumped out of these compartments without any trouble through the main drain and bilge injection suctions. A capacity test on shore was also made with one pump, and the required 5,000 gallons of water per minute were discharged under circumstances similar to those in service.

The location of the evaporating plant in the after part of engine rooms on a line with the main-engine cylinders is strongly condemned. The double-effect system on this size of vessel should be omitted, for with one evaporator in each engine room it will be difficult to work the system satisfactorily, and I fail to see the wisdom of fitting this system on any cruiser or battleship. The economy gained from its use is slight, and, if ever put to any practical use on board of a man-of-war, the extra weight and complexity is very objectionable and condemns the attachment. It certainly appears unnecessary on this vessel, for everything is

packed and jammed in the smallest space, and the operating of evaporating plant in single effect is quite enough. The evaporating plant was tested both in single and double effect.

. The two auxiliary feed pumps are located in the engine rooms while the two fire and bilge pumps are in the fire rooms. A better arrangement would have been with the feed pumps in fire rooms, and especially so, as these pumps would then be near the ash ejectors and a saving in weight of piping would have been effected.

The coal bunkers are not well arranged for rapid coaling, and the coaling chutes heavy and poorly stowed. While coaling at East Lamoine, Me., about 31 tons of coal was stowed per hour with a force of 50 stowers supplied by the contractors, and only a portion of the bunkers were filled. The question of rapid stowage of coal is one of prime importance, and should receive more consideration. It amounts to little to state how fast the coal can be handled from lighters if the coal bunkers are so situated that it is impracticable to stow the coal rapidly. (For stowage of coal see Figs. 4 and 5.)

PENALTY CLAUSE.

The penalty clause relating to the motive machinery may be a very good policy, but should be made to apply to all auxiliary machinery. Does it not seem paradoxical that there should be penalization of weights for strengthening the motive machinery when such action not only increases the endurance of the appliances but likewise gives the engine- and fire-room personnel increased confidence in the strength and reliability of the running parts, and yet permit all other departments of the ship to install appliances that are not subjected to financial penalty for overweight? Probably every part of the *Cleveland's* machinery that is overweight has been a benefit to the ship, and yet a financial penalty has been imposed for effecting such improvements, although a casual inspection of the vessel will show that weight could have been saved in other directions if all Bureaus had been subjected to the penalty that has been exacted in the case of the motive machinery.

This penalization for overweight extends to the piping, valves and to the auxiliary motive machines, while the appliances outside the engineer department are not subjected to such penalty. As a resultant of this method of penalization the contractors are naturally desirous of having those control the weights who will relieve them of penalization for putting in extra weight upon certain machines.

If it is wise policy to hold the contractors to specified weights to the machines that are subjected to the most severe strain, surely there is much more reason why auxiliary machinery throughout the ship, that is less used, should be subjected to like penalization. The question of weight penalty is one that concerns the economy, efficiency and endurance of the motive machinery, and the strong and vigorous expressions of Admiral Melville upon this subject are worthy of special consideration.

UNCOMPLETED WORK AT TIME OF INSPECTION.

1. Boiler-stop valves to be made tight.
2. Six propeller blades as spares.
3. The following packing required:
   - 2 sets metallic packing for air and circulating pumps.
   - 3 sets metallic packing for evaporator pumps.
   - 3 sets metallic packing for fire and bilge and water-service pumps.
   - 4 sets metallic packing for fire-room bilge and hotwell pumps.
   - 2 sets metallic packing for auxiliary feed pumps.
   - 1 set metallic packing for main feed pump.
4. Painting engines, piping, etc.
5. Finish covering piping.
6. Set boiler safety valves at 275 pounds.

DEFECTS IN MACHINERY.

1. Port H.P. main-engine cylinder slightly scored.
2. Port after L.P. crank pin and brasses scored.
3. Starboard H.P. and Port I.P. crosshead brasses and pins scored and ran very warm on trial.

4. Grates of boilers rise from mouth of furnace to back connection. The grates should be without any slope.
5. Engine-room spiral ladder leading from upper grating to lower grating should be changed.
6. In a great many cases where it is not absolutely necessary the stems of valves on steam piping point downward, and the stuffing boxes leak more or less. The stems should point upward whenever possible, or the stuffing boxes made deeper.
7. A lock nut should be placed on suction valve from condensers to air pumps to prevent valve closing.
8. By the present arrangement of blowers oil is thrown into the fire room and the blowers should be encased, which would materially change the blower arrangement as now fitted.
9. Instead of a large brass cover over main engine thrust bearings, separate covers should be fitted over each collar.
10. Replace metallic dial thermometers with ordinary mercurial thermometers, as the metallic thermometers do not register well.
11. Remove large lathe from workshop and fit in small gap lathe. (Large lathe 16 feet 4 inches over all.)
12. Remove grind stone and put on shore at navy yard.
13. Height of furnace doors above floor too small: difficult to fire.
14. Relief valves on feed pumps to be removed.
15. Air pumps have to be entirely dismantled from crosshead in order to examine and renew some of the valves. To renew discharge valves the head of pump has to be removed, which is quite a job.

DESIRABLE CHANGES IN MACHINERY TO WHICH THE ATTENTION OF THE INSPECTION BOARD WAS CALLED.

1. Remove evaporator plant from after part of engine room to main deck between Nos. 1 and 2 smoke pipes.
2. Counterbore steam cylinders of all auxiliary steam pumps.

3. Fit valve or valves in main steam pipes, close to throttle valves, in order to shut off engines when auxiliary steam system is in use.
4. Fit 2½-inch equalizing pipe between the upper feed tanks in each engine room.
5. Fit white metal to crosshead brasses of main engines.
6. Piston rods of auxiliary pumps, steam ends, are of composition or Tobin bronze and should be of steel, as they score badly. The main and auxiliary feed-pump rods are of steel.

RELATIVE TO INSPECTION OF MACHINERY AFTER TRIAL.

1. Open cylinders, examine pistons and balance pistons. Remove all piston valves and open up after L.P. valve of main engine.
2. Open up steam and water ends of all pumps for examination of valves and barrels. Open up valve chests of main auxiliary pumps.
3. Open for examination air pumps and valves.
4. Strip all crank pins and connect up all except H.P. and after L.P. of each engine, unless there are others in a worse condition.
5. Strip forward H.P. and after L.P. bearing brasses and roll out both bottom brasses.
6. Strip all crosshead brasses and connect up all but starboard H.P. and port I.P., unless others are found in a worse condition.
7. Thrust bearings to be opened up for inspection.
8. Remove bonnets of main condensers.
9. Open up auxiliary condensers.
10. Open up boiler drums and remove plates at bottom 4-inch tubes for inspection. Open up cleaning doors for inspection.
11. Clean bilges and paint copper pipes.
12. Pack stuffing boxes of ash-hoist engines.
13. Uptakes to be ready for inspection.
14. Fit ash-pan doors to furnaces.

15. Remove grate bars and clean out furnaces.
16. Open steam cylinders of main circulating pumps and forced-draft blowers.
17. Remove heads of feed-water heaters.

THE SMOKE NUISANCE.

During the many preliminary runs as well as on the official trial the smoke nuisance was encountered, although all possible means were taken to prevent this wastage of fuel. In war vessels, where there are all manner of limitations as to boiler installations, it is probable that the nuisance is unavoidable so long as bituminous coal is used. Smoke not only impairs the cleanliness of the vessel, but in time of war it is a notification of the presence of the ship to an opposing force.

On shore, where it is possible to build high stacks and find suitable and sufficient room for boiler installation, as well as where induced draft can be used, the smoke nuisance has been minimized.

Extended experiments should be made to determine if the evil could not be lessened on ships of war. The question of storing anthracite coal at certain naval stations might also be inquired into, for many of the fast yachts of our leading sea ports use hard coal, and but little trouble is experienced in securing maximum speed with such fuel. Tests might also be made with powdered coal, for when such fuel is used the mixture of carbon with the air might be made at such high temperatures as to produce complete combustion and thus obviate in great part the smoke nuisance. The evil will never be obliterated, however, until the fact is recognized that the boilers cannot be installed in crowded places, and that sufficient room must be allowed for efficient stoking.

THE WORK OF THE *OREGON* DURING THE SPANISH-AMERICAN WAR.

By Lieutenant C. N. Offley, U. S. N., Member.

It has been strongly urged upon the writer, from many sources, to bring more forcibly to the knowledge of all interested in naval engineering affairs the work done below the protective deck of the *Oregon* during her journey of 14,000 miles in reaching a place among the vessels of the fleet blockading Santiago. The voyage from Puget Sound to the Florida Coast was not only made in less time than was expected, but the ship reached her terminus in practically better condition than when she started on the voyage.

The principal aim of the writer in reviving this subject has been to show the importance and necessity of maintaining the motive machinery of battleships in the highest possible state of efficiency, so that in time of emergency our war vessels will be able to perform the service for which they were designed.

The work done by the *Oregon* ought to show conclusively that skill, efficiency and integrity are needful in the design, construction and installation of the motive machinery of war vessels; and that those who prepare the modern fighting machine for the day of conflict are deserving, in a large degree, of the rewards and emoluments that are tendered those who operate the warship in the hour of battle.

Faithful and efficient work had been done beneath the protective deck of the *Oregon* before she started on her eventful voyage, and if such had not been the case, the vessel would not have accomplished what she did.

From the time the *Oregon* was launched there was a determined purpose upon the part of all connected with her construction to make this vessel in all respects comparable with her

sister ships built on the Atlantic. There was an attention to detail as regards machinery design and construction that particularly manifested itself after the ship went into commission. From the hour fires were first lighted in her furnaces there was likewise a resolution upon the part of the enlisted engineer's force to secure results that would afford satisfaction to all serving on the ship, as well as to the Navy Department.

Even before the destruction of the *Maine* the efficiency of the machinery of the *Oregon* had begun to command attention. Her coal expenditure, both in port and at sea, had shown that she was as economical as any ship of her class, and her speed trials had proved that she would not have been left in the rear in the race with her sister battleships. The unexpected and welcome demand for her services on the Atlantic coast found her prepared for unusual and severe work, and therefore all who knew of her actual condition as regards motive power had no fear that she would not reach the battle line in a state of preparedness.

The greatest tribute that can be paid to officers and men who served on the *Oregon*, previous to the time that she started from Bremerton, is to simply state that when the order for the journey was received, all then serving on board believed that the ship was ready for even a more severe task. The crew of the *Oregon* well recognized the fact that this state of efficiency was due in no small degree to the work that had been done during previous years.

CHARACTER OF THE WORK DONE BY THE ENGINEER'S FORCE OF A BATTLESHIP.

It was the work that was done after the vessel started from the Pacific coast that the writer shall speak of, for a sufficient time has now elapsed since that voyage when the arduous and faithful labor performed in the engine and fire rooms should be made a matter of more prominent record. It required dogged determination, rugged endurance and a high sense of duty to labor in stuffy compartments whose temperature ranged from 120 degrees in the engine rooms to 160 degrees in the stokeholds.

There was not an hour in the discharge of this work when some unusual thump, clicking noise, hissing sound or uncertainty in the working of an appliance did not warn officer and enlisted man of some impending impairment or disaster. The thumping may have been caused by a coal bucket striking a bulkhead, and the hissing sound by some one suddenly opening some drain pipe to the bilge, but men who are at a high tension in the performance of arduous work are sensitive to slight warnings. Those in charge of the engine rooms under such conditions, or those who manipulated throttle valves, were thus under nervous and physical strain. Sleep did not come to such men immediately after they were relieved from watch, but it was often an hour or two before normal respiration and pulse permitted rest to be secured. It is, therefore, but just to those who served in the stokeholds and engine compartments, who are seldom on parade, who enter battle without the inspiration of sound of shot or shell, who work out of sight of the flag, and who are in ignorance for a time as to whether victory or defeat is imminent, that some one who knows of their discouragement and trials should unreservedly speak of their work.

ONLY A WELL-TRAINED CREW CAN PERFORM UNUSUAL SERVICE.

The performance of the *Oregon* should particularly impress naval administrators with the fact that it is as necessary to train the enlisted force serving beneath the protective deck as to train that portion of the crew whose work is in turret or redoubt. The fact should be made plain that it is as difficult work to train an efficient and reliable watertender as to train and develop any other petty officer on the warship.

There is skill and judgment in stoking that cannot be acquired during a single cruise. In the phrase of the fire room, there are on board every ship "chunkers," who can throw coal into a fire, and whose only knowledge of cleaning a fire is to haul out the live coals with the clinkers. This is the kind of firing whose resultants are leaky tubes, waste of fuel, lack of efficiency and a demoralized stokehold. Fortunately for the good of the Navy the training of petty officers for the engineer branch, as well as

the firemen, is now being carried on in a more systematic way. Such training, however, should be greatly extended, for there need be no fear that the Navy can secure too many men of the engineer's force that are able to do their work in the manner that was performed by the stokers and mechanics of the *Oregon*.

THE RESPONSIBILITY OF THE ENGINEER.

It is a military maxim that the morale of a regiment is judged by the soldierly bearing and gallantry of the field officers, and the efficiency of the enlisted personnel of the battleship is but a reflection of the manner in which duty is performed by her commanding officer and the commissioned personnel. Particularly, however does the engineer force of a war vessel follow the example set by the chief engineer. Even the title of this officer is presumptive evidence that he should keep in close touch with those serving under him. Where unusual service is performed in the engine room it is certain that the chief engineer of the ship has been the principal personage in bringing about this efficiency. One need have but little sea service to appreciate the fact that demoralization in the engine room will soon dispirit an entire ship's company. Failure in the engine room simply keeps the vessel from performing her assigned duty.

It is not the propelling engines alone which are dependent upon the efficiency of the chief engineer. While there are pneumatic, hydraulic and electric appliances under the supervision and control of other officers, all these machines are primarily operated by steam. Steam is thus the primary agency whereby the ship is steered, the anchor raised, the great guns operated and loaded. Even the ventilation and the illumination of the various compartments is dependent upon the efficiency of the boilers. The boilers are the lungs of the vessel, and where there is impairment in this direction the resultant is inefficiency of various auxiliary appliances.

The non-commissioned officers supervising the work in both engine and fire rooms should not only possess considerable mechanical skill and powers of endurance, but there must be readiness of resource, quickness of action and aptitude for the work.

A lever turned the wrong way may cause the ship to go in the direction opposite from what was intended, and a collision may be the ultimate consequence of such an error. The man without courage, steadiness of purpose and coolness in emergency will soon succumb to the strain of work in the engine room of a warship.

The fact that there can be seldom found in the engine rooms of the ocean greyhounds a gray-haired chief engineer shows that the strain and responsibility is of a severe nature, and that the pace of the work is one that kills.

COMPOSITION OF THE ENLISTED FORCE.

The difficulty experienced by the Navy Department in enlisting an adequately trained complement of watertenders, oilers and firemen ought to be conclusive testimony as to the fact that the work of the engineer's enlisted force on board ship is of an arduous and depressing nature. Comparatively few firemen from the merchant marine enlist in the Navy, although the naval service presents advantages that ought to attract young men. On board the warship there are numerous auxiliaries installed, and these appliances are worked by hydraulic, pneumatic, electrical and steam power. The Navy Regulations require frequent examination and overhauling of the various auxiliary appliances, and thus the leading petty officers of the engineer's division have excellent opportunity, during a three years' cruise, to fit themselves for better paying positions. And yet there are very few skilled firemen who will enter the service to secure such opportunities for advancement.

Duty in the engine room and fire room is severe and distressing, and it was only because the *Oregon* had an excellent complement of petty officers, firemen and coal passers that the work accomplished was so gratifying to the country.

Careful investigation shows that the average life of the fireman in the merchant service under existing boiler conditions is from seven to ten years, and it is a matter of common remark among engineers of all navies that the average age of the naval firemen is becoming less with each succeeding year.

One of the primary causes that induced the writer to revive the subject of the work of the *Oregon* was to pay a deserved tribute to the work of the enlisted force. Repeatedly between Bremerton and Jupiter Inlet there were incidents that occurred in the stokeholds that showed the courage of the man before the furnace, and it is a matter of deep personal and official regret that such acts were not made matter of official record in the steam log. The fireroom is not the place for the display of daring acts, but within those compartment walls are performed so many deeds of kindness and courage that they become too common to be made a matter of record. Men will crawl under boilers to locate a steam leak with a full realization of the fact that if their apprehensions should prove true they are crawling to their doom. In rainy and rough weather the floor plates and ladders often become so slippery that it is at the peril of one's life if he proceeds with other than extreme caution. The flanges of steam-pipe joints are often tightened to stop what appears to be a slight steam leak, and this work must often be done even when the full boiler pressure is on the pipe, although the machinist or oiler knows too well that the breaking of one of the flange bolts may result in his serious scalding. In order to respond promptly to bell signals engines are sometimes started and backed suddenly, although water in the cylinders is liable to blow out a head or bonnet. Such work, although dangerous, is of such frequent occurrence that it becomes commonplace. It shatters the nerves and health of men so employed, and one has only to be called upon to take part in such endurance runs as made by the *Columbia*, *Oregon* and *Kearsarge* to realize the permanent ill effect of being subjected to this continuous and severe strain, even for a limited period.

While some of the incidents above mentioned will apply to many ships that took part in the Spanish-American war, the writer believes that special reference to such acts forms a very important part of the work done by the *Oregon*.

MARKED JUDGMENT EXERCISED IN ANTICIPATING EVENTS.

Too much stress cannot be laid upon the fact that a most important element contributing to the success of the *Oregon* was

the care taken from the day the ship went into commission to keep every part of her machinery in the highest state of efficiency. Upon the discovery of the slightest steam leak or tendency for anything to go wrong, some one was detailed to look out for the repair. After the ship arrived in a port, however short the length of the voyage, there was a careful examination made of cylinders, valve chests, pump chambers and channel ways. No surgeon ever kept more careful watch over a patient than was given by the engineer officers to the boilers of this vessel. Unusual care was exercised that neither oil nor salt water should reach the tubes. Every leak, however small, in the boiler, came to be regarded as an excessive waste of fresh water, and thus every joint, however small and unimportant in the cycle through which the steam passed, was repeatedly examined. It was this care and vigilance that made it possible for the evaporators to produce sufficient water for general ship's purposes, as well as to make up the losses that are absolutely unpreventable about marine engines and boilers.

VALUE OF HIGH-GRADE FUEL FOR WARSHIP PURPOSES.

The character of the fuel used during the voyage was an important factor in securing gratifying results.

One must have extended experience in the operation of marine machinery to thoroughly appreciate the value of a high-grade fuel. Where there is an excellent grade of coal in the bunkers the work of the firemen not only becomes easier, but steam can be maintained more uniformly and fires can be quickly forced. There is less clogging of the tubes with soot. When the work of the firemen is eased, there is a relaxation of strain and tension that wonderfully improves the morale of the force. The men not only sleep better, but their digestion is improved. They not only become observant at such times, but they are receptive for knowledge and information, and it is no uncommon sight to see firemen upon such occasions tracing up the lead of the pipes, and coal heavers asking the privilege of cleaning and slicing fires.

The ripe experience and sound professional judgment of Chief Engineer Milligan convinced him that it was absolutely essential

that there should be within immediate reach of the firemen, for emergency purposes, several bunkers of the best procurable coal. Strict orders were given and watch was kept that these bunkers should not be touched, except in case of battle or upon some occasion where the safety of the ship depended upon the maximum power being developed. The bunkers thus reserved were always spoken of as the "fighting bunkers."

The knowledge that a large quantity of superior coal was always obtainable at a few minutes' notice by the fire-room force had a remarkable effect in convincing them that if an occasion presented itself the *Oregon* would do creditable work.

These "fighting bunkers" which were reserved for emergency purposes were the athwartship ones. The firemen thus realized that if a call to battle should be sounded the wing bunkers would be tightly sealed, and thus the coal would serve as a protection. The men also knew that with these bunkers closed the chances of the ship being sunk by being rammed by an enemy was lessened, since water entering through the opening would be confined to the bunker compartments.

THE CONDITION OF THE HULL.

As the vessel had come out of dock just before she took her departure from Bremerton, the hull was free from barnacles and grasses. Her outboard valves had been carefully examined, and everything found in excellent order as regards the line shafting. Practically every examination and repair that had been done to engine and boilers was carried on by the engineer's force of the ship, and thus the crew had an intimate knowledge as to the condition of things. There was not only an efficient but a contented organization, and this was a resultant of keeping the men steadily at work, but taking special care that there should be neither nagging nor harassing upon the part of any superior.

THE EARLY DAYS OF THE VOYAGE.

The voyage should be considered as starting from Bremerton. The crew well understood that the Department regarded it as highly probable that the vessel would be called upon to engage

in battle, otherwise there would have been no call for a battleship to make her way to the Cuban coast from Puget Sound. The ship remained ten days in San Francisco filling bunkers and magazines, and taking aboard such stores and spare parts as might be required on a protracted blockade or for use in action. On March 19, 1898, she took her departure from the Golden Gate, heading direct for Callao and proceeding under three boilers. The Peruvian port was reached on the morning of April 4th, the *Oregon* having consumed during this run of sixteen days, 900 tons of coal, leaving 600 tons still in the bunkers.

The run from San Francisco to Callao was an efficient performance, the vessel having been steamed 4.24 knots per ton of coal. The revolutions of the engines were remarkably steady, an average speed of 75 turns being maintained day after day without a variation of a tenth of a revolution in engine speed.

On two occasions during this run slight leaks began to appear around the back ends of some of the tubes in one of the boilers. A reserve boiler was immediately put in use, and fires allowed to die out in the slightly impaired boiler. Just as soon as it was possible for the boilermaker to commence operations the leaky tubes were re-expanded. A most searching examination was always made for any impairments or minor defects, and it was not many hours before the furnaces of the boiler that had been leaking were primed. The watertenders kept as close watch as the commissioned officers that the water in the boilers should be kept fresh. There was a daily output from the evaporators and distillers of about 5,000 gallons of fresh water, and this was sufficient for all purposes, including the necessary make-up feed for the boilers.

FIRE DISCOVERED IN ONE OF THE COAL BUNKERS.

On the afternoon of March 27th, when about half way between San Francisco and Callao, smoke was discovered issuing from one of the coal bunkers. This bunker held about 15 tons of coal, and at the time was about half full. It may seem a simple matter in such a small compartment to locate the exact position of the fire, but one has only to encounter such an ex-

perience to note how difficult it is to reach the burning portions. By reason of the size and nature of the compartment only two men could work at a time, and they were relieved every ten minutes. It is extremely dangerous work to fight such a fire, for the danger from suffocation or from inhaling highly heated gases is very great. Although only about a shovel full of live coal was discovered, there were several tons of coal that had become so hot that smoke and hot gas was issuing from that fuel. It required about four hours of steady work to remove all the dangerous coal, and in this operation assistance from the deck force was secured. The usual speed of the ship was maintained while the fire was being fought. The fire brigade that cleared the burning bunker was in charge of Naval Cadet Jenson, and only those who were on board the vessel and knew of the risk that attended the entrance to a bunker compartment which had become excessively heated, and which was filled with poisonous gases, could get a conception of the danger which Jenson and his force were in.

THE STAY AT CALLAO.

During the stay at Callao necessary minor repairs were made to the starboard and port engines. The brasses were scraped where necessary; main cylinders and valve chests examined and oiled, and air pumps cleaned and repaired. In the port engine room one of the main crosshead slippers was found to be so badly cut that it was deemed advisable to remove it and replace it with a spare one that was carried on board; this repair required thirty-six hours of continuous work, as the face had to be scraped to a smooth surface and the guides carefully adjusted.

While important work was being done in the engine room, there was overhauling where necessary in the fire room. The boilermaker and his force went through the furnaces and examined tubes and back connections. The main drainage system was overhauled and tested, and there was a most thorough cleaning of the bilges. The blowing engines were examined where necessary, and firing tools put in order.

Under the supervision of the Cadet Engineers Jenson and Leahy, a large contingent of firemen and coal passers were en-

gaged in trimming bunkers. Each bunker was personally inspected by the young officer in charge of the coaling. Coal was actually stowed into the spaces between the deck beams. All hands realized that it was not only a matter of doubt where the ship would coal again, but there was an apprehension that the next invoice of coal might be of inferior quality, and therefore it was of primary importance that every possible pound of good coal should be taken on board. There is no harder work than stowing the bunkers of a warship, for these compartments are over and around the boilers, and this trimming work is very exhausting. At all the principal coaling ports of the world the "trimmers" receive double the pay that is given those who carry on board the coal. If it is hard work for "trimmers" to stow the bunkers of merchant ships that will average three times the size of the bunkers of the warship, then it is easy to understand why the close stowing of bunkers on a battleship is a work that uses men up. The stowing of the bunkers was always done under the supervision of the two cadets named.

THE RUN FROM CALLAO TO PORT TAMAR.

When the ship left Callao, April 7th, the belief was general on board the vessel that war with Spain was inevitable. At the start three boilers were in use, the vessel averaging something over 11 knots per hour. On the evening of April 9th, a fourth boiler was connected, which increased the speed to 13 knots, and this speed was maintained until the evening of the 16th, when Port Tamar was reached.

On the run from Callao one or two tubes had been found leaking in one of the boilers, but these leaks were soon stopped. On this run also, in some manner which has never been determined, a small amount of salt water got into the boilers, about just enough to cause the density of the water to become one-quarter that of the sea water. This slight feeding of salt water into the boilers proved exceedingly advantageous, since it caused the formation of a layer of scale thick enough to close all leaks at the ends of the tubes, yet so thin that there was no perceptible loss of efficiency. That small feed supply of salt water to the

boilers caused all in the fire room to be more vigilant than ever before that the valves to the sea should be carefully watched.

During the night of April 16th–17th the vessel lay at anchor in Port Tamar. As the weather was bad, heavy fires were kept in all boilers. The engines were kept in readiness to work to bell signal, and the ship was prepared to slip anchor at a moment's notice.

THE PASSAGE THROUGH THE STRAITS.

On the morning of April 17th, the voyage was resumed. Captain Clark had expressed a great desire to reach Sandy Point before dark, but was not inclined to order the engineer's force to make a day's run under forced-draft conditions, by reason of the severe character of such work. Both officers and enlisted force, however, of the engineer's division volunteered to go into forced-draft watches, and therefore the run that day was made under assisted draft. This considerate action of the commanding officer of the ship appealed very strongly to firemen and coal passers, and it was such manner of kindly sympathy and encouragement that inspired in the men a high sense of duty. During the run through the Straits the blowers were run at a speed that gave an air pressure in the fire room of about a quarter of an inch of water. The engines were run at a speed of about 107 revolutions per minute, which gave the ship a speed through the water of over 14½ knots per hour. Sandy Point was thus reached in good time, and there was a feeling of intense satisfaction throughout the ship that on the next run each day would lessen the distance to the probable battle ground.

THE SPIRIT OF OFFICERS AND MEN.

The unusual strain was beginning to make itself felt, but the hope of reaching the Cuban coast in time to take part in a possible Trafalgar gave strength and encouragement to every one on the ship, whether his duty was in turret or bunker, or whether his work was behind a gun or in front of a furnace.

It had been a month since the ship left San Francisco, and there were days in the intervening period when it seemed as if the vessel was too far away from the probable scene of action to

reach the battle ground in time. It was then that all hands believed that their labor would be for naught, and that they would reach the Cuban coast just in time to help escort the victorious fleet of Admiral Sampson to a home port. It was then also that officers had to dismiss their own forebodings, and to give assurance that the Commander-in-Chief of our forces would wait for the *Oregon* before attacking Havana, or some other strongly fortified port.

To quote from an article previously printed in the JOURNAL: "On more than one occasion, when the labors of the men before the furnaces seemed to go for naught, and they felt all but discouraged because the steam would not 'go up,' the engineer officers, discarding their uniforms, jumped among them, fed the furnaces with their own hands, and by force of this example so encouraged the firemen, that, with cheers, they redoubled their efforts, and the good ship sped through the water at a faster pace."

FROM SANDY POINT TO KEY WEST.

The first run on the trip northward was to Rio de Janeiro, which port was reached April 30th. It is regrettable that the limited space of a magazine article will not permit the telling of incidents that took place on the several runs, but which showed the keen desire of all on board the vessel to reach the battle line in time. Upon reaching the Brazilian port news was received that war had been declared. The ship only remained long enough in Rio to take on coal, and while receiving this fuel some necessary overhauling was done. By reason of the difficulty of getting coal, her stay in Rio was much longer than anticipated.

On May 5th the vessel left Rio and reached Bahia three days later, an average speed of a little over 10 knots having been made on this run. The coal received at Rio was only of a fair quality, and thus the work of the firemen and coal passers became more arduous.

On May 10th the ship left Bahia for Barbadoes, and reached that port on the 17th. During the voyage from Rio to Bahia, and from Bahia to Barbadoes, a sharp lookout was kept for any

Spanish ships, and therefore things were kept in a state of preparedness for action. It is needless to state that steaming under such conditions throws a strain and responsibility upon men that is not undergone when only cruising conditions exist.

During the stay at Barbadoes 240 tons of coal were taken on board. On May 19th the vessel started for Jupiter Inlet, Florida, which point was reached May 24th. Her arrival having been reported to the Navy Department, Captain Clark was directed to proceed to Hampton Roads in case the ship was in need of repairs, otherwise to steam to Key West. There was no hesitation as to which direction to take under these instructions, and therefore on the morning of May 25th the *Oregon* began her journey to Key West, where she arrived the following day.

ORDERED TO JOIN A BLOCKADING SQUADRON.

In writing of the performance of the U. S. S. *Oregon*, the late Lieutenant C. A. E. King, U. S. N., states: "Ordinarily, after a run of more than 14,500 knots, a great many repairs would be found absolutely necessary even to a cruiser, and several weeks would not be considered too long a time to spend in rehabilitating the engine and boilers of any ship after such extraordinary service. But the *Oregon* coaled, rapidly completed a few adjustments of machinery, repaired a few minor defects and at 1:04 on the morning of May 29th, less than three days after arrival in that port, steamed away from Key West to add her strength to that of our fleet off Santiago de Cuba."

Captain Clark, upon reporting to Admiral Sampson at Santiago, declared the *Oregon* ready for any service. The joy and satisfaction of joining the blockading fleet in time to assist the other battleships in shelling the forts or in attacking Cervera's squadron made all hands forget the weary watches on the voyage from the Pacific coast. There was thus a noticeable improvement in the bearing and action of the men, and there was an intuitive feeling that in some manner it would be shown before the war was over that the long journey of the *Oregon* would not be fruitless in results.

DUTY WITH THE BLOCKADING FLEET.

During the five weeks of blockade duty the *Oregon* was ready at all times for efficient service. Her readiness for battle was shown upon several occasions. At one time she received a signal which was interpreted to read, "Report name of steamer six miles to southward." Seeing the smoke of a steamer in the distance the *Oregon* was headed for the strange vessel. In a few minutes the forced-draft blowers were working full speed, steam was up to the limit, and, the ship was speeding through the water. After an hour's run word was sent down to the engine room that the vessel chased was a large three-masted ocean steamer, and apparently a prize worth taking. By that time the engines were turning 125 revolutions per minute, with a consequent hot journal. The engine speed, however, was not lessened by a fraction of a turn. The oilers and spare machinists used water, oil and the usual cooling substances, and within fifteen minutes the brasses had cooled to a blood heat. Within another half hour the steamer was overhauled, and it was found that the capture was one of the fast newspaper dispatch vessels, a boat which up to that time had believed that she could out-speed any ship of the blockading squadron. The ship on this run averaged over 16½ knots, but to bring about this speed both officers and men of the engineers' force were very much exhausted. This unexpected call for maximum speed conclusively showed that everything in regard to the boilers and engines was in the highest state of efficiency.

HIGH PROFESSIONAL QUALITIES OF THE OREGON'S CHIEF ENGINEER.

The condition in which the *Oregon* reached the blockading fleet, and her subsequent performance in battle as regards machinery is the highest tribute that can possibly be paid to the signal ability of Chief Engineer R. W. Milligan. Many incidents could be told of his faithful and efficient performance of duty, and of the high esteem and admiration with which he was regarded by the enlisted force.

The following incident will show the high professional judgment of the man, and of his intuitive knowledge of what was

required of a blockading ship. One of the boilers had started a slight leak in the tube ends and back connections. As usual, the defects were repaired immediately. Upon reporting the completion of the work, the Chief Engineer requested permission of the Captain to relight fires in the boilers, but was told that in the Commanding Officer's opinion it would be unnecessary to carry steam on all boilers during blockade work. Chief Engineer Milligan, however, asked that he might use his own judgment in the matter. The request having been granted by Captain Clark, the order was at once given to start the fires and connect this boiler with the others as soon as the pressures in all boilers was the same. Succeeding events sustained the judgment of the Chief Engineer, since the exceptional speed made by the *Oregon* during the chase of the Spanish squadron was only made possible because the vessel had steam on all boilers.

Another order of the Chief Engineer which proved of great service in quickly getting the ship under full speed, was that the oil cups on the forced-draft blowers should always be kept filled and the wicks kept in the cups. The Chief Engineer realized that under the excitement of going into action there might be danger that some of these oil cups might be overlooked. Furthermore, a great saving of time was effected at a period when there were other important things for the petty officers to do.

ESTEEM AND AFFECTION WITH WHICH THE CHIEF ENGINEER WAS REGARDED.

In telling of the work of the *Oregon* it becomes the writer's pleasure and duty to tell of the exceeding regard in which Chief Engineer Milligan was held by his subordinate officers, as well as by the enlisted force serving under him.

Four years after the voyage of the *Oregon*, and when deep reflection and calm judgment had superseded hysterical sentiment, there was presented to Captain Milligan by the commissioned officers and cadet engineers who served with him during the Spanish American war, a loving cup. The inscription on this testimonial best tells the story of the admiration and esteem that the junior officers had for their chief engineer.

On one shield of this cup are the following words:

"To Commander R. W. Milligan, U. S. Navy, Chief Engineer of the U. S. S. *Oregon*, in appreciation of the distinguished service rendered to the Navy and to the Nation by him on board that vessel during the Spanish-American war."

On the reverse shield are engraved these words:

"This token is a testimonial of the high courage, professional esteem, military regard and personal affection in which Commander Milligan is held by the junior officers of the engine department who had the honor of serving under him before and during the battle of Santiago."

THE VOYAGE OF THE *OREGON* DISTINCTIVELY AN ENGINEERING TRIUMPH.

Excellent service was performed by every officer and enlisted man on board the *Oregon* in bringing the ship from Bremerton to the blockading fleet. The journey of 14,000 miles was distinctively a test of the machinery design and installation, and of the *personnel* whose privilege it was to operate the plant on that memorable voyage.

That the Congress recognized the voyage as an engineering triumph was evidenced by the fact that not only the Chief Engineer, but every commissioned officer doing duty in the engine room was advanced numbers for the work done on the trip and at Santiago. Reference to this matter is only made because the writer desires to make known the surprise and disappointment of Chief Engineer Milligan and his commissioned subordinates, that Naval Cadets H. N. Jenson and W. D. Leahy, of the engineer division, were not likewise given special promotion. Only those who made the voyage can also have a realization of the efficient work that was done in bringing the vessel to the Cuban coast by the Navigating Officer, now Commander Reginald F. Nicholson. All on the vessel hoped that promotion would have been also extended him.

Cadets Jenson and Leahy stood regular watches in the engine rooms from the day the *Oregon* left Puget Sound until they were detached from the vessel. At the several ports at which the vessel stopped, they not only entered the boilers, but they saw

to the stowing of the coal, and in many respects their work was more arduous than that of the commissioned engineer officers. They would enter the furnaces at the earliest possible time after the fires were hauled, and upon several occasions the bridge walls were so hot that it was only by covering the brick work with boards and burlap that an examination of the tubes, tube sheets and back connections could be made by these young officers. It is hoped that what was undoubtedly an oversight as regards the failure to substantially recognize the work of these young officers may soon be corrected, for Captain Clark accorded them the same testimonials that he gave all other commissioned officers doing duty in the engine rooms.

THE BATTLE OF SANTIAGO.

On July 3d, 1898, when the the first of Admiral Cervera's ships was seen coming out of harbor, it was but a few minutes before the fires in the furnaces were spread and gun turrets were in readiness for operation. Ten minutes later the ship was steaming under full forced draft, and before the close of the day, a speed of nearly 16 knots was attained.

As to the work of the *Oregon* during that day, the following quotations from official reports will best tell of her performance:

"The *Oregon*, having proved vastly faster than the other battleships, * * * continued westward in pursuit of the *Colon*." (Official report of Commodore W. S. Schley, U. S. N., upon the battle of July 3, 1898.)

"The *Oregon* was keeping up a steady fire and was coming up in a most glorious and gallant style, outstripping all others. It was an ispiring sight to see this battleship, with a large white wave before her, and her smokestacks belching forth continued puffs from her forced draft." (Official report of Captain F. A. Cook, U. S. N., commanding the *Brooklyn*, upon the battle of July 3, 1898.)

"The *Oregon*, steaming with amazing speed from the commencement of the action, took first place. The fine speed of the *Oregon* enabled her to take a front position in the chase, and the *Cristobal Colon* did not give up until the *Oregon* had thrown a

thirteen-inch shell beyond her. This performance adds to the already brilliant record of this fine battleship, and speaks highly of the skill and care with which her admirable efficiency has been maintained during a service unprecedented in the history of vessels of her class." (Official report of Rear Admiral W. T. Sampson, U. S. N., upon the battle of July 3, 1898.)

THE SERVICE RENDERED THE NATION BY THE COMMANDING OFFICER OF THE *OREGON*.

The world has always and will ever associate one particular man in connection with every great event, and therefore the story of the battleship *Oregon* will be the story of the Captain of the vessel, now Rear Admiral C. E. Clark, one of the most modest, unassuming and bravest of able sea captains.

In a signal manner Admiral Clark has been honored by his native State as well as by the Navy Department, the Congress and the country at large. He is a hero to those who served under him on the *Oregon*, and these men saw him manifest great qualities upon many occasions.

On the roll of naval heroes his name will stand for professional efficiency, high moral worth and courage, and for those attributes which must be possessed by leaders who are called upon for the performance of great deeds.

THE NICLAUSSE BOILER.

By Charles M. Howe, Member.

There seems to be a wide diversity of opinion on the subject of the Niclausse boiler as a successful type for marine use. On the one hand, and mainly because of the mishaps occurring on the U. S. S. *Maine*, the boiler has been severely criticised in this country; on the other hand, it is favorably regarded in other countries, notably in England, in which country the committee appointed by Parliament to investigate the question of boilers for the British Navy recommend the Niclausse among the four types to be tried. In view of these contrary opinions, and because of the popular interest attaching to a subject of such vital importance to the United States Navy, it may not be inopportune to present a brief description of the Niclausse boiler as designed and built in this country by The Stirling Company.

That a type of boiler which has been in existence only ten years should have come into the general use indicated by installations aggregating seven hundred thousand horsepower in the ships of twelve different navies at once suggests that the boiler possesses some features of merit. It is also worthy of note that the Niclausse boiler has succeeded in gaining a foothold in the mercantile marine, a number of ships abroad being equipped with this type, in addition to the 33,000-ton Great Northern ships, *Minnesota* and *Dakota*, building in this country. Furthermore, it is the only type of marine water-tube boiler that has yet to record a failure, no Niclausse boiler ever having been removed from any ship.

The boiler was developed by Messrs. J. & A. Niclausse, of France, and in the United States is built by The Stirling Company, of Chicago, who, while adhering in the main to the principles of Messrs. Niclausse, have made radical modifications and improvements in design; and the American built type may properly be referred to as the "Improved" Niclausse boiler.

DESCRIPTION.

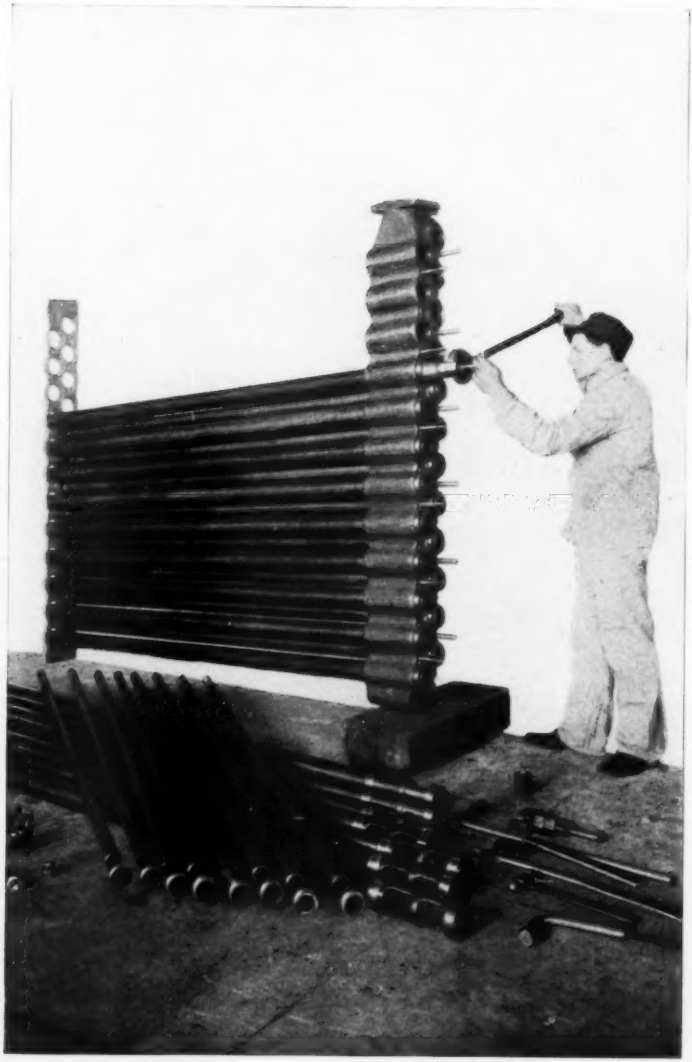
Generally the boiler may be said to consist of a number of separate "elements," each of which comprises a vertical header to which the tubes are connected. These elements are surmounted by a transverse drum; into which the headers lead, and the whole boiler is enclosed by the usual sheet-steel casing.

TUBES.

The boiler utilizes the principle of the "Field" tube, in that the outer or generating tubes are closed at one end and contain inner or circulating tubes, whose only office, as may be inferred, is to provide for circulation. The means of attaching these tubes to the headers constitutes perhaps the most unique, and certainly the most distinguishing characteristic of the Niclausse boiler. Each evaporating tube is provided at one end with two cones, distant apart about eight inches, the first of which fits into a coned recess in the front face of the header, and the second in the rear face. This is the "Niclausse joint." To the engineer, who has always associated ideas of boiler-making with tubes expanded into a tube-sheet this means of making a joint which shall be steam-tight and safe does not appeal at first; but the fact remains that it is not only possible to make such a joint steam and water-tight, but it is actually done in the seven-hundred thousand horsepower Niclausse boilers now in use, and the most rigid tests have failed to develop even a sign of leakage where the joint is made.

The generating tubes are made of steel, and are seamless-drawn. The earlier construction employed a malleable cast-iron lantern for the coned bearings, but this has been succeeded by an entirely new model in which the lantern as a separate piece has been discarded, and one end of the tube itself is shaped to perform the functions of the lantern. This is done by turning one cone on an upset extremity of the tube proper, and the second cone on an expanded portion thereof. Located between the two cones is another expanded portion, but this is not a bearing surface and merely serves to occupy the tube-hole in the diaphragm in order to render the latter effective in dividing the two currents.

REMOVING A CIRCULATING TUBE.



REMOVING A GENERATING TUBE.

By cutting away diametrically opposite portions of the tube the form of the original lantern is closely approximated, and, although the tube is now entirely in one piece, the term "lantern" is still applied to that portion of it which lies within the header.

The circulating tubes are made from thin sheet steel. They are subject to no pressure and are made as light as is consistent with the necessary rigidity. At one end they carry the circulating-tube lantern, by means of which the tube is attached to the end of the evaporating tube and is held in place concentric with the latter.

HEADERS.

The construction of the headers for the boilers has perhaps demanded more engineering skill and entailed the expenditure of more money than any single feature of any boiler ever before placed on the market. Originally the headers were made of malleable cast iron and were corrugated or sinuous on their lateral faces. The demand, however, was for a wrought-steel product, adapted to the high steam pressures then coming into use. Messrs. Niclausse met this demand by a rectangular header made from pressed steel, but the United States manufacturers went a step further, and produced a sinuous header of seamless steel, resembling in every way the old cast-iron form and retaining the advantages of that form, but made from a single piece of seamless tubing. In this header, as developed by The Stirling Company, and its commercial production obtained only after the exercise of the greatest ingenuity and the design of special methods and machinery, the purchaser of the Niclausse boiler acquires a specimen of boiler construction unequaled anywhere for strength and mechanical perfection; and the demand for something better than cast iron between human life and 300 pounds pressure has been successfully met. The construction of these headers involves not only the difficult problem of inserting a dividing diaphragm into the sinuous member, but the equally delicate operation of flanging the tube holes outward, in order to present the necessary surface for the coned recesses of the Niclausse joint, all of which necessarily must be done with great precision. The lower ends of the headers are closed, and the upper ends are swaged down for connection to the overhead drum.

STEAM AND WATER DRUM.

This is of the usual construction, and is made in conformity with the best prevailing high-pressure practice, all plate being of the best grade of low-carbon open-hearth steel, with rivet holes drilled and all rivets hydraulically driven.

In the earlier forms of the boiler, as made by both Messrs. Niclausse and The Stirling Company, the drums were 31½ inches diameter, but in the improved type, as designed for the U. S. S. *Pennsylvania*, *Colorado*, *Virginia* and *Georgia*, and for the *Minnesota* and *Dakota*, the diameter has been increased to 42 inches, which has been adopted as the standard size and is used whenever the available headroom permits. The advantages are greater steam and water space, and greater accessibility. The drums in the first installations were somewhat completely occupied by various devices for the precipitation of salts, regulation of feed, etc., many of which have been dispensed with by the United States manufacturers, in the interests of rendering the boiler more simple and the interior of the drum more accessible for examination and cleaning.

CASINGS AND ACCESSORIES.

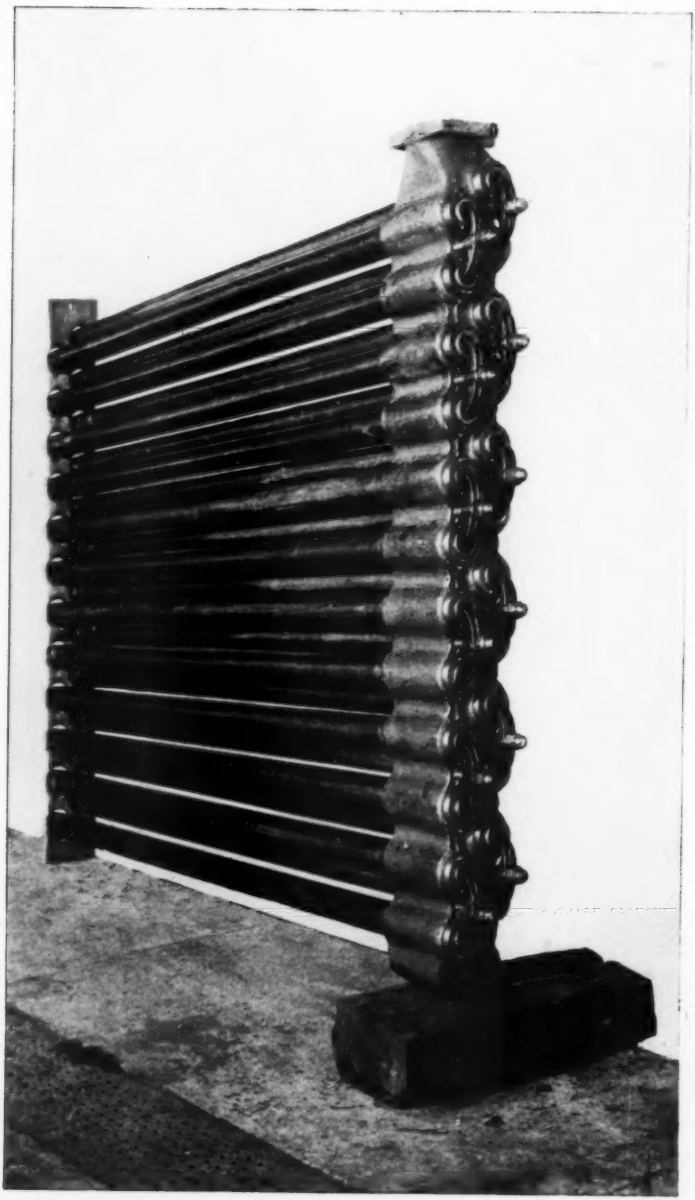
Of these little need be said. The casings are sheet steel lined with asbestos and fire brick, and the boiler is fitted with the usual safety, steam and blow-off valves, etc. The firing and ash-pit doors are of the balanced type, opening inwardly.

CIRCULATION.

The circulation of water in the Niclausse boiler is along well-defined paths and is particularly active. Positive and rapid circulation is essential in any boiler supposed to steam properly, and the Niclausse admirably fills every requirement in that respect.

The headers are divided into a front and a rear compartment by the vertical diaphragm. The inner or circulating tubes terminate at this diaphragm and have a water connection only with the front compartment. In a similar manner the evaporating tubes connect only with the rear compartment. The header diaphragm is prolonged through the coned nipples connecting the headers and the drum, and thus the paths for the descending and ascending currents are positively defined. Water from the drum descends the front compartment of the headers, flows into

NICLAUSSE BOILERS FOR THE GREAT NORTHERN STEAMSHIP COMPANY'S "MINNESOTA." ELEMENTS UNDERGOING HYDRAULIC TEST AT THE WORKS OF THE STIRLING COMPANY, BARBERTON, OHIO.



A Niclausse Element.

the inlets of the circulating tubes, along the length of these tubes to their rear and open ends, reverses direction and starts back through the annular space between the tubes, is evaporated in transit, enters the rear compartments of the headers and ascends to the drum above. At once there is afforded a positively defined path, short, and free from contractions and obstructions of any kind—assuredly all the *desiderata* of a system wherein circulation is to be created and maintained by natural causes. That the circulation can be retarded by forcing the boiler beyond the limits for which it was designed is well known, but where undue forcing can be anticipated by the makers this contingency can now be met.

CAPACITY.

The question of the features of design operating to limit the capacity of the Niclausse boiler has been definitely solved by The Stirling Company, after tests and experiments ranging over several years, and the entire matter may be said to turn upon the proportions of the rear compartment of the header—the up-channel or steam passage. The Stirling Company have investigated this point with great care and painstaking, and as a result of experiments, which subsequent tests have convincingly established the accuracy of, have obtained a definite ratio between the sectional area of the up-channel of the headers and the limiting rate of fuel consumption, in consequence of which the design of headers for the Great Northern ships and the U. S. S. *Pennsylvania*, *Georgia*, *Colorado* and *Virginia* is materially different from those in the earlier Niclausse installations made by the company, and the improved type of header will admit of a combustion rate far in excess of anything likely to be attained under the most extreme conditions of forced draft.

ADVANTAGES OF THE NICLAUSSE BOILER.

1. Its circulation is positive, rapid and free. It follows a well defined, short path, and one free from restricted passages or other obstructions.

2. It is easy to repair. The facility with which those members most subject to wear—the tubes—can be removed and replaced is unequaled, unapproached, by any boiler extant. To withdraw any tube and replace it with another is a matter in-

volving a few minutes' time, and is in marked contrast with the elaborate procedure required for the same operation in other types of boilers. Mr. William A. Fairburn, in his paper entitled "The Water-Tube Boiler in the American Mercantile Marine," read before the Society of Naval Architects and Marine Engineers, at the meeting in November, 1902, presents this contrast very forcibly, somewhat as follows:

To remove a tube from any double-header type of boiler:

First. Remove the cap over the tube at the front end.

Second. Remove the cap over the tube at the rear end.

Third. Split the end of the tube in its front end seat, so that the part that projects through the seat can be closed in or reduced in diameter, so that it will pass through the tube hole.

Fourth. Repeat this operation at the rear end.

Fifth. Drive the tube through the header with the aid of dolly balls and sledges until it projects through the front header far enough to attach a sling to it.

Sixth. Connect a set of blocks in the front end of the tube and haul it out, provided the tube is not blistered or bulged in the center. If such is the case, then the front end must be cut and the tube driven back and dropped into the fire box; and probably other lower tubes will have to be cut out before the damaged tube can be removed, thus greatly increasing the time, cost and number of operations required in making the desired repairs.

It will be noted that the defective tube must be split previous to closing in the end, and to make this splitting effective it must be carried inward beyond the tube seat. Would any competent engineer think of intrusting this splitting of the tube to a common laborer when the risk of grooving the seat would be so great?

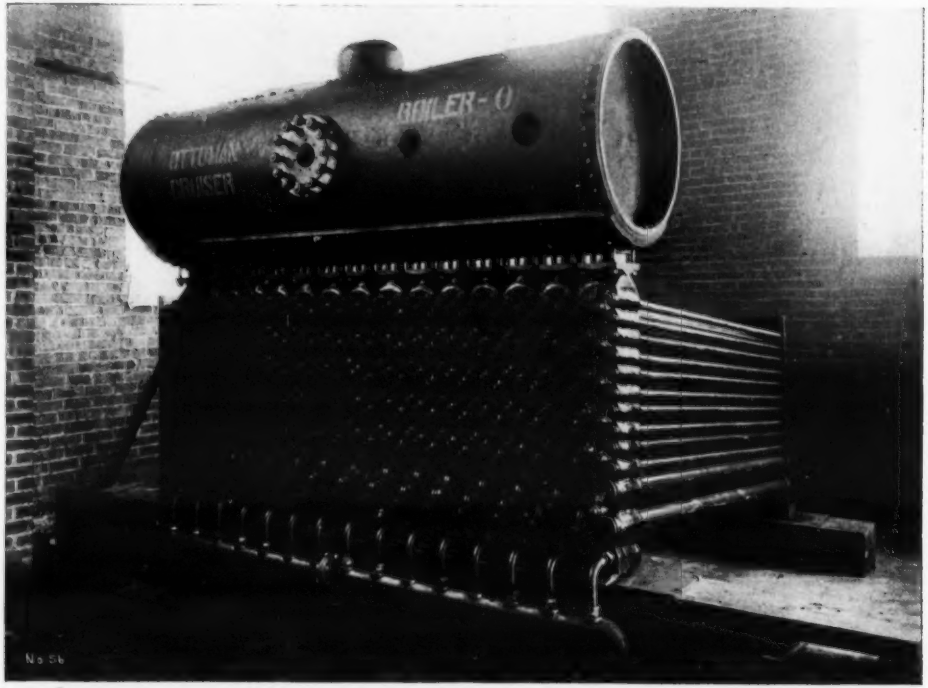
In the event that the tube is not split, then the end must be closed in by special tools or bars. This cannot be accomplished through the tube-cap hole in the front of the face of the header, unless the edge of the hole is used as the fulcrum of the lever, and the effect on this hole and on the seat for the closing cap can be imagined.

To remove a tube from a Niclausse boiler:

First. Remove the circulating tube.

Second. Insert tube puller and remove generating tube. If the tube is badly blistered it will be cut off as described in the case of the double-header boiler. If the damaged tube should be in the second or third row from the bottom, the tubes below it can readily be withdrawn to permit the removal of the blistered portion, and this without the sacrifice of any additional tubes.

FORWARD GROUP OF SIX NICLAUSSE BOILERS, IMPERIAL RUSSIAN CRUISER "VARIAG," SHOWING COMBINED END AND SIDE-BATTERY ARRANGEMENT.



NICLAUSSE BOILERS FOR THE OTTOMAN CRUISER "MEDJIDIA," WITH CASING REMOVED.

It is claimed by Messrs. J. & A. Niclausse that on the Russian gunboat *Khrabry*, with the ship under way at full power, steam was lowered, the boiler emptied, a tube changed, the boiler refilled and pressure raised again—the whole operation occupying not more than thirty-five minutes.

3. It is easy to clean, can be kept clean, and therefore can be operated at a maintained rate of economy. The exterior of the tubes in the boiler may be cleaned from the boiler's front, whereas in other types the tubes are accessible for cleaning from the sides only. From an operative point of view this is one of the strongest points of merit in the Niclausse boiler, for it means that, whatever the size of the boiler units, the economic results obtained at first can be obtained always, provided the fire-room force avails itself of the excellent cleaning facilities offered.

4. It has no cast parts subject to pressure, but, on the other hand, is a production of wrought steel throughout. Even the cast-iron lanterns formerly used, although not subject to pressure in the slightest degree, have been abandoned in favor of the single-piece evaporating tube.

5. The Stirling Company's design includes an elevation of the tube bank sufficiently far removed from the grates to provide an ample combustion chamber, insuring the proper mixture of the gases after distillation and their ultimate thorough combustion.

6. The boiler is absolutely free for expansion. The tubes are fixed at one extremity only, and merely rest in supports at the other, through which supports they are perfectly free to move. No boiler made presents less resistance to expansion.

7. It is very economical of space. Primarily, the projected area and the volume of the Niclausse boiler are less per thousand square feet of heating surface than any other type, with the possible exception of the Belleville. But an additional consideration of the greatest importance is the fact that, because the boiler is cleaned from the front and does not require space at the sides for this operation, nor room at the rear for the removal of tubes, tube caps, etc., the space ordinarily reserved for these purposes may be utilized to better advantage.

DATA FROM OFFICIAL TRIALS OF NICLAUSSE BOILERS.

Names of ships.	Date.	Duration of trial. Hours.	Coal per sq. ft. of grate per hour.	Draft.	Evaporation per lb. of coal, from and at 212°.	I.H.P.	Coal per I.H.P. per hr.	Speed. Knots.
Léon Gambetta ..	Feb. 10,03	6	22.53	F.	11.061	...	...	...
Léon Gambetta ..	Feb. 14,03	3	34.82	F.	10.706	...	...	...
Léon Gambetta ..	Feb. 19,03	6	16.38	N.	11.680	...	...	...
Gueydon	Nov. 28,02	6	13.53	N.	...	1,923	1.58	9.627
Gueydon	Dec. 9,02	6	19.66	N.	...	13,754	1.74	18.747
Gueydon	Dec. 23,02	3	32.15	F.	...	19,395	1.95	20.30
Gueydon	Jan. 30-31,03	24	15.41	N.	...	10,828	1.64	17.920
Suffren.....	Aug. 5,02	6	20.48	F.	10.415	...	...	...
Suffren.....	Aug. 8,02	4	26.63	F.	10.730	...	...	...
Suffren.....	May 23,02	6	12.92	N.	...	1,936	1.90	...
Condé.....	Feb. 1,01	6	15.36	N.	12.091	...	...	...
Condé.....	Feb. 5,01	6	22.53	F.	11.992	...	...	...
Condé.....	Feb. 18,01	3	34.82	F.	10.965	...	...	...
Marceau	Feb. 5,03	6	18.43	N.	11.205	...	...	...
Marceau	Feb. 10,03	4	32.77	F.	10.204	...	...	...
Henri IV	Apr. 18,02	6	9.22	N.	11.685	...	...	...
Henri IV	Apr. 22,02	6	18.43	N.	10.354	...	...	...
Henri IV	Apr. 25,02	6	26.63	F.	10.204	...	...	...
Henri IV	Apr. 29,02	4	35.84	F.	9.004	...	...	...
Henri IV	Apr. 28,03	6	11.45	N.	...	1,773	1.62	10.023
Henri IV	May 13,03	6	12.17	N.	...	1,747	1.75	10.183
Henri IV	May 19,03	24	13.92	N.	...	6,017	1.75	15.008
Henri IV	June 5,03	6	17.55	F.	...	7,639	1.75	15.918
Friant	Dec. 12,94	6	10.24	N.	...	3,607	1.49	15.36
Friant	Dec. 21,94	6	11.48	N.	...	1,602	1.61	11.73
Friant	Dec. 28,94	6	18.26	N.	...	7,091	1.92	17.975
Friant	Mar. 15,95	4	25.00	F.	...	9,310	...	18.891
Friant	Apr. 19-20,95	24	16.36	N.	...	6,194	1.87	17.099
Davout.....	June 20,01	4	32.77	F.	10.080	...	...	...
Davout.....	June 29,01	6	18.43	N.	10.935	...	...	...
Davout.....	Oct. 10,02	4	31.41	F.	...	7,911	...	18.68
Requin.....	Jan. 14,02	6	18.87	F.	...	5,926	1.89	15.332
Requin.....	Jan. 21-22,02	22	14.70	N.	...	4,708	1.76	...
Fleurus	Feb. 3,99	6	15.83	N.	...	1,058	2.05	13.341
Fleurus	Feb. 15-16,99	24	14.60	N.	...	2,060	1.94	15.15
Fleurus	Feb. 25,99	3	33.92	F.	...	3,967	...	17.7
Temeraire.....	Sept. 16,98	2	40.96	F.	11.282	...	...	...
Temeraire.....	Sept. 17,98	2	51.20	F.	10.820	...	...	...
Temeraire.....	Sept. 19,98	3	66.56	F.	10.728	...	...	...
Temeraire.....	Sept. 21,98	3	81.93	F.	10.300	...	...	...
Décidée	May 25,00	6	...	...	...	334	1.54	...
Décidée	June 00	24	...	...	...	543	1.71	...
Décidée	June 00	6	...	...	...	903	1.83	...

F. = forced. N. = natural.

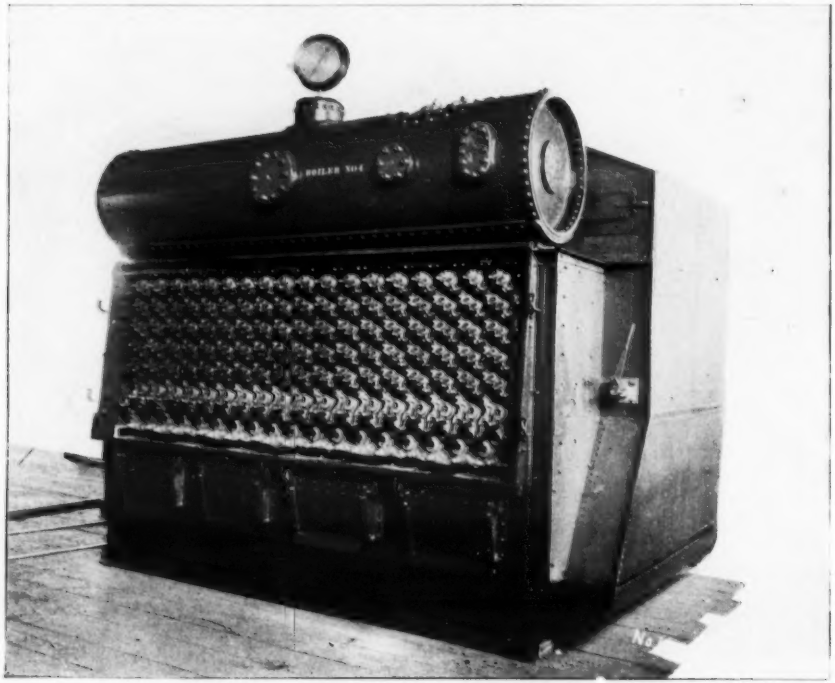
DATA FROM OFFICIAL TRIALS OF NICLAUSSE BOILERS.—Continued.

Names of ships.	Date.	Duration. Hours.	Coal per sq. ft. of grate per hour.	Draft.	Evaporation per lb. of coal, from and at 212°.	I.H.P.	Coal per I.H.P. per hour.	Speed. Knots.
Zélée	June 16,00	6	9.75	N.	...	332	1.28	10.201
Zélée	June 19,00	6	17.48	N.	...	940	1.62	13.315
Zélée	June 22-23,00	24	10.55	N.	...	576	1.58	10.040
Hercule	Dec. 24,98	8	20.48	F.	9.020	...	...	...
Titan	May 30,00	6	10.24	N.	...	291	1.48	...
Titan	.....	4	...	F.	...	477	1.22	...
Polypheme	July 13,00	6	...	F.	...	534	1.13	...
Polypheme	July 17,00	6	...	N.	...	315	1.25	...
Elan	Aug. 26,91	6	19.30	N.	9.970	...	...	...
Elan	Aug. 27,91	4	32.15	F.	8.990	...	...	...
Elan	Sept. 1,91	6	17.15	N.	10.220	...	...	...
Elan	June 4,96	6	31.54	F.	10.035	...	...	...
Retvizan	Oct. 24,01	2	9.00	N.	...	8,186	1.67	...
Khrabry	Sept. 98	2	...	...	...	2,170	1.20	...
Khrabry	Sept. 98	2	...	...	...	2,170	1.47	...
Khrabry	Sept. 98	4	...	...	...	2,170	1.88	...
Cristobal Colon	Apr. 10,98	6	...	N.	...	4.695	...	15.
Cristobal Colon	Apr. 13,98	6	...	N.	...	4.537	...	14.8
Cristobal Colon	Apr. 28,98	4	...	N.	...	10,553	...	19.33
Cristobal Colon	Apr. 30,98	4	..	N.	...	11,343	1.79	19.60
Cristobal Colon	May 4.98	8	...	N.	...	8,596	1.65	18.26
Berwick	Feb. 18,02	8	27.65	F.	10.544	...	...	...
Berwick	Feb. 19,02	8	16.38	N.	11.823	...	...	...
Berwick	Feb. 21,02	8	22.53	F.	11.537	...	...	...
Berwick	June 03	30	...	...	...	4,612	1.76	14.85
Berwick	June 03	30	...	...	...	16,326	1.81	21.644
Berwick	June 03	8	...	...	...	22,370	1.93	23.613
Seagull	Jan. 17-20,00	61	17.04	N.	...	1,609	1.87	14.6
Seagull	Feb. 2,00	56	13.76	N.	...	1,772	1.87	15.2
Seagull	Mar. 2,00	65	18.23	N.	...	1,774	1.85	15.38
Seagull	Mar. 16-18,00	53	20.70	N.	...	1,921	1.87	16.7
Fantôme	Jan. 10,01	30	...	...	...	334	1.72	9.25
Fantôme	Jan. 13,01	30	...	...	...	1,006	1.55	12.5
Fantôme	Jan. 16,01	8	...	...	...	1,433	1.55	13.5
Garibaldi	Aug. 17,01	6	...	N.	...	9,812	1.90	...
Garibaldi	Nov. 22-23,01	24	17.79	N.	...	6,722	1.92	17.05
Freya	Mar. 26-27,01	24	...	...	...	1,184	1.68	...
Freya	Mar. 29-30,01	24	...	...	...	5,918	1.74	...
Gazelle	Feb. 99	6	30.72	F.	...	5,918	2.15	...
Gazelle	Feb. 99	24	...	...	...	3,187	1.79	...
Gazelle	Feb. 99	8	...	...	...	1,184	1.65	...
Yaeyama	June 17,01	2	14.34	N.	11.640	...	...	...
Niitaka	Oct. 20,02	4	22.53	F.	10.230	...	...	..
Niitaka	Oct. 25,02	4	22.53	F.	10.250	...	...	...
Niitaka	Oct. 27,02	1	34.82	F.	9.920	...	...	...
Tsushima	Feb. 18,03	2	18.43	F.	11.820	...	...	...

F. = forced. N. = natural.

NICLAUSSE INSTALLATIONS.

Name of ship.	Type.	Displacement.	No. of boilers.	Grate surface.	Heating surface.
				Sq ft.	*Sq. ft.*
	FRANCE.				
Patrie	Battleship	16,385	24	1,378	45,994
République	Battleship	16,385	24	1,378	45,994
Léon Gambetta	Armored cruiser	13,889	28	1,661	55,682
Condé	Armored cruiser	11,020	28	1,187	39,956
Gloire	Armored cruiser	11,020	28	1,187	39,956
Gueydon	Armored cruiser	10,472	28	1,215	39,525
Kléber	Armored cruiser	8,488	20	1,099	35,637
Suffren	Battleship	14,030	24	1,137	41,912
Marceau	Battleship	11,994	16	916	32,240
Henri IV	Battleship	9,865	12	807	25,937
Jean Bart	Cruiser	4,916	12	603	19,870
Isly	Cruiser	4,916	12	603	19,870
Friant	Cruiser	4,133	20	783	23,247
Davout	Cruiser	3,337	10	669	21,140
Requin	Battleship	7,716	12	603	19,325
Fleurus	Torpedo cruiser	1,433	8	269	8,181
Temeraire	Torpedo boat	192	2	73	3,229
Décidée	Gunboat	716	2	86	2,174
Zélée	Gunboat	716	2	87	2,705
Travailleur	Tug	...	1	49	1,589
Hercule	Tug	...	1	49	1,589
Utile	Tug	...	2	70	2,210
Titan	Tug	360	1	45	1,467
Polypheme	Tug	360	1	45	1,467
Elan	Pilot schoolship	265	2	43	1,356
Agile	Torpedo boat	135	2	49	2,172
	RUSSIA.				
Variag	Cruiser	7,165	30	1,600	63,109
Retvizan	Battleship	14,000	24	1,346	58,125
Khrabry	Gunboat	1,984	8	224	6,781
Ocean	School ship	13,228	6	258	8,267
	SPAIN.				
Christobal-Colon	Cruiser	7,716	24	957	30,964
Pelayo	Battleship	10,968	16	853	28,149
	ITALY.				
Regina Margharita	Battleship	14,000	28	1,386	44,506
Garibaldi	Armored cruiser	8,157	24	1,098	35,216
Francesco Ferruccio	Armored cruiser	8,157	24	1,098	35,216
	ENGLAND.				
New Zealand	Battleship	18,022	...	...	...
Berwick	Armored cruiser	10,803	34	...	52,312
Suffolk	Armored cruiser	10,803	34	...	52,312
Devonshire	Armored cruiser	11,240	...	...	...
Carnavon	Armored cruiser	11,240	...	...	...
Seagull	Gunboat	810	6	262	7,556
Fantôme	Gunboat	1,200	4	135	3,993
Cadmus	Gunboat	1,200	4	135	3,993
Clio	Gunboat	1,200	4	135	3,993



ONE OF THE NICLAUSSE BOILERS FOR THE U. S. S. "NEVADA."

NICLAUSSE INSTALLATIONS.—Continued.

Name of ship.	Type.	Displacement.	No. of boilers.	Grate Surface.	Heating surface.
				Sq. ft.	*Sq. ft.*
GERMANY.					
Freya	Cruiser	6,228	12	807	26,403
Gazelle	Cruiser	2,866	8	...	15,629
UNITED STATES.					
Colorado	Armored cruiser	15,873	32	1,484	65,436
Pennsylvania	Armored cruiser	15,873	32	1,484	65,436
Virginia	Battleship	18,408	24	1,280	56,526
Georgia	Battleship	18,408	24	1,280	56,526
Maine	Battleship	13,780	24	1,345	58,125
Nevada	Monitor	3,566	4	220	8,874
TURKEY.					
Messoud'je	Battleship	10,041	16	842	26,985
Abdul-Mejid	Cruiser	3,582	...	...	...
JAPAN.					
Yaeyama	Cruiser	1,764	8	357	10,841
Niitaka	Cruiser	3,770	16	754	23,491
Tshushima	Cruiser	3,770	16	754	23,491
ARGENTINE REPUBLIC.					
Presidente Sacmiento	Schoolship	3,031	2	...	...
PORTUGAL.					
Saint-Gabriel	Cruiser	1,984	1	5.5	144
Patria	Cruiser	...	...	2.76	75
CHILI.					
Esmeralda	Cruiser	8,267	1	...	1,078

8. It is economical of weight. It is extremely light *per se* and in contained water, and yet these *desiderata* have been attained without any sacrifice of either strength or safety from low water.

9. It has no screwed joints in contact with the fire. The only screwed joints anywhere near the fire in the boiler are those at the extremities of the evaporating tubes, where conically-threaded caps are employed, and these are amply protected from all action of the fire by the complete baffling effect of the cast-iron tube rests.

Other points might be mentioned in the boiler's favor, but the foregoing will suffice to show that it is not inherently the defective type of generator that some would insist. Its advantages are generally recognized in other countries, and this fact is of the utmost importance, because the opinions there formed have

been based less upon theoretical considerations than upon the results of *actual application in a large number of large installations;* and assuredly the use of any single type of boiler to the extent indicated in the attached table should at least suggest that the boiler is not without merit. This table, as well as that of the data from official trials, relates only to Navy ships, and does not include the very considerable number of Niclausse boilers in the mercantile marine and in stationary plants.

NOTE ON THE DIRECTION OF ROTATION OF PROPELLERS ON TWIN-SCREW WAR VESSELS.

By M. Laubeuf, Chief Constructor, French Navy.

Translated by Chief Engineer F. G. McKean, U. S. N. (Retired), "Bulletin de L'Association Technique Maritime," No. 14, Session de 1903. Paris, 1903.

The question as to which direction it is advisable to make twin screws turn has been in debate ever since the beginning of the use of two screws. I will quote on this subject a note by M. Huin,* Director of Naval Construction (retired), at present Director-General of the "Chantiers de la Gironde," the conclusion of which is that, viewed from the stern, the starboard screw should turn in the direction of the hands of a watch, the port screw in the opposite direction, to give the best results. For simplicity's sake, we will give to this arrangement the number 1 (screws diverging above), and we will call that in which the propellers turn in the other direction arrangement No. 2 (screws converging above).

I believe that M. Normand, our eminent Vice-President, is also an adherent of the arrangement No. 1 (screws diverging above). At any rate, vessels issuing from his shops all have this direction of rotation. M. Garnier, Director of Naval Construction at Toulon, formerly Sub-Director of the establishment at Indret, is also in favor of this direction of rotation.

On the other hand, M. Doyère, Chief Constructor, French Navy, whose works on propellers are well known to the members of the Technical Maritime Association, upholds the arrangement No. 2 (screws converging above), and M. Sigaudy, chief engineer of the Société des Forges et Chantiers de la Méditerranée, at Havre, has also adopted that arrangement, No. 2 (screws converging above), for quite a number of engines.

* "Mémorial du Genie Maritime," 3e Livraison, 1887, Ass. Tech. Mar., 1903.

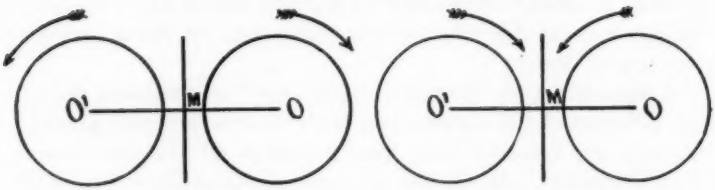
The question, therefore, is not yet thoroughly decided. It is for this reason I venture to return to it.

Furthermore, here follows the direction of rotation adopted in various twin-screw vessels:

Arrangement No. 1 (screws diverging above).	Battleships.	*Hoche, Magenta, Marceau, Neptune, Amiral-Baudin, Dévastation, Courbet, Amiral-Duperré, Brennus, Charles-Martel, Jaurèguiberry.*
	Armored Cruisers.	*Latouche-Tréville, Bruix, Charner, Chanzy, Pothuau.*
	Cruisers.	*Davout, Suchet, Friant, Bugeaud, Chasseloup-Laubat, du Chayla, Pascal, Descartes, Alger, Isly, Jean-Bart, Catinat, Protet, Tage, Cécille, Linois, Galilée, Lavoisier, Surcouf, Forbin, Troude, Lalande, Cosmao.*
	Destroyers.	*Cassini, Casabianca, Condor, Faucon, Vautour, Épervier, Fleurus, Watignies, Durandal, Hallebarde, Fauconneau, Espingole, Léger, Lévrier.*
Arrangement No. 2 (screws converging above).*	Battleships.	*Formidable, Carnot.*
	Cruisers.	*Cassard, d'Assas, Coëtlogon.*
	Destroyers.	*D'Iberville, Dunois, Lahire, Pique, Épée, Yatagan.*

Fig. 1.—Stern View.

LOOKING FROM THE REAR TOWARD THE STERN.



Arrangement No. 1. Arrangement No. 2.

The reason which, in late years, has caused the adoption of arrangement No. 2 (screws converging above) on several vessels, is that, quite wrongly, in my opinion, it is the engine builders who decide upon the screws. The propeller, in fact, as M. Doyère has remarked, is bound up intimately with the hull of

* As stated in the last paragraph of this article, the following triple-screw vessels in the French Navy have their outside or wing screws turning inboard: *Jeanne d'Arc, Montcalm, Gueydon, Dupetit-Thouars, Sully, Marseillaise*, etc.—EDITOR.

the ship which it drives, and ought to be studied by the builder of the hull. As to the engine builder, it suffices to give him an outline of a propeller which permits the absorption of the estimated power at the given number of revolutions.

Now, the adoption of the direction of rotation of the arrangement No. 2 (screws converging above) has, in the eyes of engine builders, a very seductive advantage, it enables the crosshead guides to be placed outboard, and, consequently, in plain sight of the attendant, who usually occupies a passage situated towards the center of the vessel.

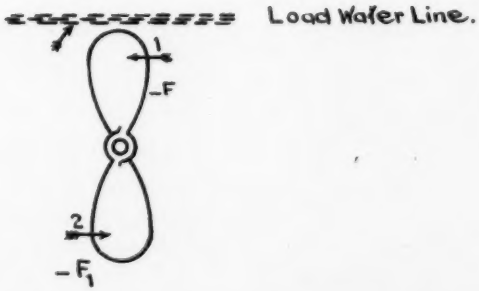
This is the only advantage which I admit for this arrangement. We think it must yield precedence to the advantages of arrangement No. 1 (screws diverging above). They are of two kinds:

1. ADVANTAGES IN MANEUVERING.

It is known that with a single screw turning in the direction of the hands of a watch, as viewed from astern, a ship falls off to port.

The explanation of this is simple. Let us consider (Fig. 2) one blade.

Fig. 2.



When it is vertical, as at 1, it exerts on the water an oblique thrust, of which the component situated in the plane perpendicular to the shaft, has a certain value F; the reaction on the blade is $-F$, equal and of the opposite sign.

When the blade is vertical, as at 2, it works in a more resisting medium, and the reaction of the water — F_1 is greater than F. There results, therefore, from this a pushing of the stern of the vessel to starboard, that is to say that the vessel falls off to port.

M. Normand* has explained the same fact by the inclination of the stream elements, reaching the screw in an ascending direction. It results from this that the thrust is greater for the descending blade than for the ascending one. Therefore, with the single screw turning in the definite direction above indicated, the blade passing horizontally to starboard gives a greater thrust than the blade passing to port, whence results the falling off of the ship to port.

I think that the two explanations are valid, and that the two effects indicated both exist.

With two screws, when we have the arrangement No. 1 (screws diverging above), if in order to turn more rapidly we stop the port one, the starboard screw tends to cause falling off to port: 1st, because of the couple, *thrust* $\times$ *OM* (Fig. 1); 2d, because of its own action by the effect indicated above.

With arrangement No. 2 (screws converging above), on the contrary, if we stop the port engine, the starboard screw tends to cause falling off to port, because of the couple, *thrust* $\times$ *OM, but that effect is opposed by the action of the screw itself*, which would tend to make the vessel fall off to starboard.

From the maneuvering point of view there is, therefore, no doubt that the propellers ought to turn according to arrangement No. 1 (screws diverging above), which diminishes the radius of the vessel's turning circle.

2. ADVANTAGES OF EFFICIENCY.

As we have said above, M. Huin finds that the advantage in this respect remains with the arrangement No. 1 (screws diverging above).

Here is another demonstration of that proposition based, like

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* "Mémorial du Genie Maritime," 7e Livraison, 1883.

the argument of M. Huin, upon the inclination of the stream line in relation to the horizontal and longitudinal axis.

1st. *Effect of the inclination of the stream lines in the horizontal plane.*

If we trace on the bottom of a ship several horizontal sec-

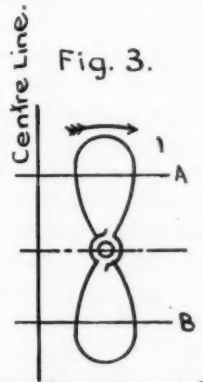
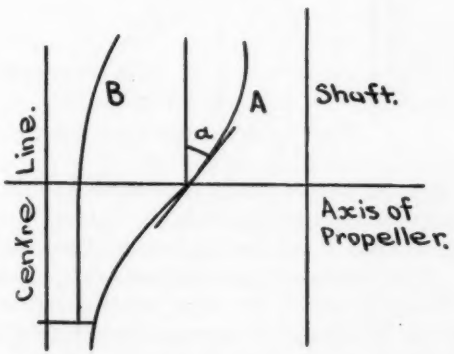


Fig. 3.

Vertical projection viewed from astern.

Fig. 4.



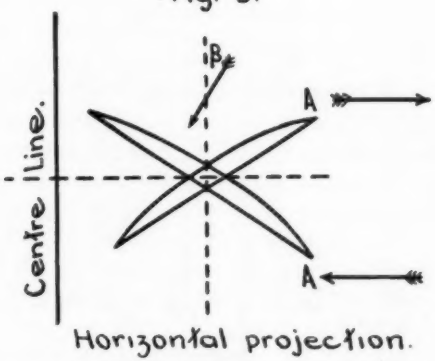
Horizontal projection.

tions cutting supposedly vertical propeller blades, for example section *A* and section *B* (Fig. 3) at ¾ of the radius, we note that, while section *B* (Fig. 4) is, on all ships, parallel to the axis at a point opposite to the screw, section *A*, on the contrary, deviates perceptibly from this parallelism.

The angle α is 5 degrees 30 minutes on the *Jean-Bart* and *Isly*; 6 degrees 30 minutes on the *Alger*; 7 degrees 15 minutes on the *Bugeaud*, *Friant*, *Chasseloup-Laubat*; 3 degrees on the *Cécille*; 4 degrees 45 minutes on the *Pothuau*; 7 degrees on the *du Chayla*, *Cassard*, *d'Assas*, *Latouche-Tréville*, *Bruix*, *Charner*, *Chanzy*; 2 degrees 30 minutes on the *Galilée* and *Lavoisier*.

Consequently the stream lines striking the blade which is situated at *B* parallel to the axis, they strike the blade situated at *A* at an angle which may deviate several degrees distant from the preceding (Fig. 5).

Fig. 5.



Let us suppose a screw designed according to the Drzewiecki method, with an angle of approach of 2 degrees 51 minutes.

The lower blade *B* has that angle indeed, but the upper blade has not. If the starboard propeller turns in the direction No. 1 (screws diverging above) the upper blade *A* acts as if it had a greater pitch. The angle of approach is increased by the angle β made by the stream lines and the horizontal plane.

On the contrary, if the starboard propeller turns in the direc-

tion No. 2 (screws converging above), the upper blade A^1 has an angle of approach diminished by β.

Now, it results from various trials which we have made with screws, of which the angle of approach varied from 1 degree to 8 degrees, that the efficiency of the screw varies but little when that angle varies between 1 degree 45 minutes and 5 degrees, but while it decreases rather slowly above 5 degrees it decreases very rapidly below 1 degree 45 minutes.

If, then, the stream lines reach the upper blade at an incidence of $\beta = 2$ degrees only, which is certainly exceeded in vessels of medium fineness, the angle of approach becomes 4 degrees 52 minutes for the direction of rotation No. 1 (screws diverging above), and 0 degrees 52 minutes for the direction No. 2 (screws converging above). The blade will have a fair efficiency in the first case, a very moderate one in the second.

If the angle of incidence reaches 3 degrees, the angle of approach of the upper blade is about 6 degrees in the arrangement No. 1 (screws diverging above), 0 degrees in the arrangement No. 2 (screws converging above). The upper blade no longer exerts any thrust in this latter case.

Finally, if the angle of incidence exceeds 3 degrees the upper blade works in the opposite way while turning in the direction No. 2 (screws converging above), while its efficiency is simply diminished a trifle in the direction No. 1 (screws diverging above).

Besides, the upper blade working in a less resisting medium, there is advantage only to be gained by increasing its angle of approach. Consequently, the direction of rotation No. 1 (screws diverging above) is the better.

2d. *Inclination of the stream lines in the vertical plane.*

If we pass vertical sections through different points of blades supposedly horizontal, we see that the angle of these sections with the horizontal plane, in fine vessels, is very nearly the same throughout the horizontal projection of the cylinder of water set in motion.

Propellers being supposed to be set suitably distant from the hull, we may, consequently, admit that the currents of water impinge on all points of the circle described by the screw with

equal speed and at an equal angle of incidence with the horizontal.

Under these conditions, we will see, as above, that the angle of approach of the descending blade is increased, that of the ascending one diminished. In other words, the thrust of the descending blade is increased, that of the ascending blade is diminished.

The disturbing effect produced is the same, whichever may be the direction of rotation adopted; it is therefore unimportant from this point of view.

Results of experience.—It remains for us to see if the results of experience confirm the theory set forth.

In the first place, all who have been present at high-speed trials on vessels in which the second arrangement (screws converging above) is adopted, have been surprised to see that there was formed at the stern in the axis a liquid ridge, a kind of crest much raised, standing almost vertically, like a wall, at a considerable height. This height exceeded 6.5 feet in the *Dunois* and *Lahire*. It seems as if the liquid masses hurled by each screw are compressed against each other.

It is certain to any unprejudiced mind that the stern-wave formed thus is much more considerable than with the arrangement No. 1 (screws diverging above), in which, on the contrary, the current of water from the propellers appears to disperse.

There is, therefore, greater loss of energy by the formation of waves with arrangement No. 2 (screws converging above), and, consequently, a less efficiency than with arrangement No. 1 (screws diverging above).

Finally, by means of the trials of several vessels we can draw comparisons which appear to us entirely conclusive on this point.

These vessels are the *Durandal* and the *Hallebarde* on one side, the *Pique* and the *Épée* on the other.

The hulls are absolutely identical, and have been built on the plans of M. Normand. But, while the engines of the *Durandal* and *Hallebarde*, constructed by M. Normand, turn as in arrangement No. 1 (screws diverging above), those of the *Pique* and *Épée*, built by the Société des Forges et Chantiers, at

Havre, on the plans of M. Sigaudy, turn in the direction No. 2 (screws converging above).

Here follow the data of the propellers, and the results of the trials, at nearly equal powers.

The data are those of the screws in position for the final trials (two sets of screws were tried on the *Hallebarde*, four sets on the *Pique*):

Propellers.	*Durandal.*	*Hallebarde.*	*Pique.*	*Épée.*
Number of blades	4	4	4	4
Outside diameter	2^m, 31	2^m, 34	2^m, 30	2^m, 30
Pitch :				
At the radius of 0^m, 60	3^m, 085	3^m, 065	2^m, 96	Has not been stated. Increasing like that of the *Pique*.
At the radius of 0^m, 90	3^m, 165	3^m, 145	3^m, 025	
At the radius of 1^m, 10	3^m, 260	3^m, 234	3^m, 08	
Mean	3^m, 192	3^m, 170	3^m, 0216	3^m, 0282
Fraction of pitch in blade :				
At the radius of 0^m, 60	0^m, 1322	0^m, 1342	0^m, 1455	0^m, 1385
At the radius of 0^m, 90	0^m, 0937	0^m, 1013	0^m, 1075	0^m, 1047
At the radius of 1^m, 10	0^m, 0474	0^m, 0545	0^m, 0487	0^m, 0475
Mean	0^m, 0911	0^m, 0967	0^m, 10056	0^m, 0969
Diameter of hub	0^m, 32	0^m, 32	0^m, 32	0^m , 32
Results of the trial at full speed :				
Number of revolutions	301′, 61	297′, 67	300., 4	300′, 97
I.H.P.	5350 (*)	5199$^{ch.v}$	5441$^{ch.v}$	5429$^{ch.v}$
Speed in knots	27, 415	26^n, 9	25^n, 887	25^n, 621
Immersed midship section B^2	7^m, 987	7^m, 894	7^m, 97	8^m, 075
Displacement D	279′, 949	276′, 091	287^t, 2	297′, 23
Efficiency M in $V = M\sqrt[3]{\frac{F}{B^2}}\cdots$	3, 13	3, 092	2, 94	2, 92
Effic'ncy M^1 in $V = M^1\sqrt[3]{\frac{F}{D^{\frac{2}{3}}}}\cdots$	5, 482	5, 415	5, 20	5, 167

* Following the curves resulting from the progressive trials.

So, here are two vessels identical as to hull. The screws are very slightly different; the hubs have the same diameter. It is very probable also that the engines have an efficiency differing little; the displacement is, it is true, a little greater in the *Pique* and *Épée*.* But we believe that neither the difference of immer-

*Concerning the relative performance of similar machinery of similar vessels under different conditions of operation, it is only necessary to note the performance of the *Oregon* with that of the *Indiana*. The official Navy Register shows that both ships are of 10,288 tons displacement. Their engines are in all respects similar, both having been designed, built and installed under the direction of engineer officers of the naval service, and yet the horsepower of the *Indiana* is 9,738 while that of the *Oregon* is 11,111. There

sion, nor the difference of the screws, nor the difference of the engines suffices to explain a difference of efficiency such that at 5,400 I.H.P. there is a deviation in speed from 27.25 knots (the mean for the *Durandal* and *Hallebarde*), to 25.75 knots (the mean for the *Pique* and *Épée*), namely, 1.5 knots, an enormous difference at these speeds. †

We conclude, therefore, that arrangement No. 1 (screws diverging above) is more advantageous than arrangement No. 2 (screws converging above). We should be allowed further, in order to eliminate nearly all the causes which vitiate the comparisons, to make the following test: The *Pique* should make a series of runs at different speeds in her present condition, then she should go into dock. They should fit the starboard propeller to port and inversely, and the vessel should make a new series of runs at the same number of revolutions, *the engines backing*.

As for vessels with three screws, I willingly admit that from the point of view of efficiency, the side screws are far enough from the hull and far enough from each other to permit the adoption of either direction of rotation. But the diminution of the turning circle gained by arrangement No. 1 (screws diverging above) remains, and that advantage ought to suffice to cause the adoption of that arrangement for the side propellers of triple-screw vessels.

I note on the contrary, with regret, that on all our recent vessels with three screws (*Jeanne d'Arc*, *Montcalm*, *Gueydon*, *Dupetit-Thouars*, *Sully*, *Marsellaise*, etc.), they have adopted the arrangement of screws converging above.

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is at least 2 knots difference in the average sea speed that can be maintained by the two ships, and these facts show that some surprising results can be secured by reason of superior workmanship and operation in the case even of similar vessels. Other illustrations could be given from foreign navies as to the marked difference of performances of similar ships, thus showing that the question of personnel is a very important factor in the attempt to secure comparative experimental data.—THE EDITOR.

† We will draw attention, in passing, to the greatness of the circumferential speeds realized. They amount to 119.75 feet [per second] which shows that that of 116.47 feet, which we have indicated for the propellers that should be adopted on the *Dunois* and *Lahire* ("Bulletin de l'Association Technique Maritime," 1901) was not exaggerated.

U. S. BATTLESHIP *MISSOURI.*

DESCRIPTION OF VESSEL—OFFICIAL TRIAL PERFORMANCE.

By Lieutenant Commander F. C. Bieg, U. S. N., Member.

The *Missouri* is a twin-screw, armored battleship, built by the Newport News Shipbuilding and Dry Dock Co., on their own plans for the machinery. The contract was signed January 7, 1899, the keel laid February 7, 1900, and the hull launched on December 28, 1901. The launching weight was 4,907 tons, including 315 tons for cradle and shoring. The contract price was $2,885,000, of which $860,000 was for machinery. This price does not include the armor and armor bolts (exclusive of the requirements of the protective deck), the ordnance and ordnance outfit, and certain articles, as anchors and chains, supplied by the Government, but does include the fitting and installing of the above.

The time allowed for completion was thirty-two months, but this was afterwards extended seventeen months. A further extension was made necessary in 1903, when the ship was nearly completed, on account of changes made in the supports and rollers for the turrets. The collective indicated horsepower of propelling, air-pump and circulating-pump engines was to be 16,000, with the main engines making about 120 revolutions per minute.

The speed guaranteed was 18 knots, to be maintained successfully for four consecutive hours, the air pressure in the fire rooms not to exceed an average of two and one-half inches of water, the vessel to be weighted to a mean draught of 23 feet 6 inches. If the speed fell below 18 knots, the contract inflicted a penalty at the rate of $25,000 per quarter knot for speeds between 18 and 17½ knots, and $50,000 per quarter knot for speeds between 17½ and 17 knots. In case the speed fell below

17 knots, it was to be optional with the Secretary of the Navy to reject the vessel or accept her at a reduced price.

The complement will be 35 officers and 612 men.

HULL.

Length between perpendiculars, feet	388
on L.W.L., feet	388
over all, feet and inches	393- 9½
Beam, extreme, feet and inches	72- 2½
at L.W.L., feet and inches	72- 2½
Ratio of length to beam	5.44
Depth (molded), feet and inches, main deck	34- 5⅞
upper deck	42- 6½
Height, flying bridge above main deck, feet and inches	32- 6
Draught, normal, mean, feet and inches	23- 6
Displacement, mean load draught (23 ft. 10½ in.), tons	12,460
normal draught (23 ft. 6 in.), tons	12,240
full load draught (25 ft. 2½ in.), tons	13,278
per inch at L.W.L., tons	50.35
Area midship section to L.W.L., square feet	1,618
L.W.L. plane (23 ft. 6 in.), square feet	21,130
Wetted surface, square feet	36,700
Center of gravity of L.W.L. plane, aft of M.S., feet	2.65
buoyancy above bottom of keel, feet and inches	12–10½
forward of M.S., feet	1.2
gravity above bottom of keel, feet	25.24
Transverse metacenter above center of buoyancy, feet	17
Longitudinal metacenter above center of buoyancy, ft. and ins.	408- 6
Coefficient of fineness block	.653
of M.S.	.956
L.W.L.	.756
Cylindrical coefficient	.686
Number of frames	98
watertight compartments	283

The hull is of basic open-hearth steel, of domestic manufacture, with frames spaced 4 feet apart, except where intermediate frames are required, from the stem to frame 81; from this point aft the frames are spaced 3½ feet.

Of the 283 watertight compartments, 12 are in the double bottom. The double bottom proper extends from frame 30 to frame 71, and consists of compartments B–91 to 99 and C–97 to 99, B–96 and 97 being reserve water tanks. Forward and aft of

these points, or throughout the limits of the magazines, the bottom plating of the magazines forms a continuation of the inner bottom; in the shaft alleys, the plating continues to frame 80. A cofferdam, 36 inches wide, is constructed between the berth and protective decks on each side of the vessel, extending from the diagonal armor bulkheads to the ends of the vessel.

The cofferdams are filled with fireproofed, corn-pith cellulose, made in suitable briquettes and compressed to a density of from 8 to 9 pounds per cubic foot. All wood used in the construction of the vessel, except the teak armor backing, is fireproofed, and the use of wood is avoided as far as possible. The upper decks and main deck, outside of the superstructure, are laid with wood; the others are covered with linoleum.

There are two military masts, each with two tops for machine guns. The bridge deck extends between the turrets and forms the upper part of the superstructure. On it are the boats and 3-inch and 1-pounder guns.

Upper Deck.—This extends from the forward side of the after turret to the stem. There are four 6-inch guns aft on this deck, protected by 6-inch armor. Abaft of these guns is a suite of rooms for the executive officer, the latter's office, the navigator's and some junior officers' rooms, and the mess room for the latter. Forward are the officers' and crew's galleys. Two winches are fitted on this deck.

As in most modern ships, the engine-room hatches, small in area, end on this deck, and, being covered by the bridge deck above and enclosed by the superstructure, their utility as air ducts is somewhat impared, to say the least.

Main Deck.—On this deck, inside of the casemate, is located the rest of the 6-inch battery in broadside, with splinter bulkheads between the guns. On this deck are located also the crew's water closets and showers, the windlass engine, ordnance and pay offices, drying rooms, junior and warrant officers' quarters, berthing space for part of the crew, and a winch aft of the turret.

Berth Deck.—Here are the quarters of the admiral, captain and officers, the wardroom, dining room, the engineer's office or

logroom, a dark place built in a coal bunker, the engineer's workshop, berthing space for part of the crew, the firemen's washrooms, machinists' and petty officers' quarters, the refrigerating plant, prison, sick bay, dispensary, blower rooms, lavatories and coal bunkers above the protective deck.

Protective Deck.—The space between the berth and protective decks is given up to stores.

Splinter and Platform Decks and Hold.—Below the protective deck are the machinery spaces, dynamo rooms, coal bunkers, holds, chain lockers, trimming tanks, torpedo and storerooms, magazines, shell rooms, two blower rooms, steering gear, air compressors and ammunition hoists.

Drainage System.—The main drain pipe is of galvanized steel, 14 inches in diameter and runs from the after part of the forward boiler compartment on the port side to the after part of the port engine room, above the double bottom, 14-inch branches leading to each engine and boiler compartment and to each main circulating pump. It is fitted with stop check valves in each boiler and engine compartment, all worked from the berth deck. The suction pipes leading from these valves in the boiler rooms do not dip into drainage wells, but have their mouths located as near the inner bottom as practicable and covered with coarse strainers. The main drain pipe also connects with the principal steam pumps other than the feed and water-service pumps.

For the auxiliary drainage system, manifolds are fitted in convenient locations throughout the machinery space, with suction pipes leading to the principal lower compartments of the ship, double-bottom compartments, reserve and trimming tanks, with connections to the sea as required. Where these pipes drain water from compartments above the double bottom, a small well is fitted, projecting 6 inches below the inner bottom and covered by a plate strainer. All manifold valves are worked only from the manifolds. A 5-inch drain pipe, above the double bottom, connects the after and forward manifolds with flanges for pump connections. In the boiler and engine compartments, the suctions are fitted with flanges for a direct connection to the pumps in that compartment. The crank pits are drained independently

of the rest of the bilge. The compartments above the protective deck are drained by scuppers or portable pumps. The coal bunkers below the protective deck are drained by their own vertical sliding doors.

ARMOR.

The armor consists of a side belt of Harveyized nickel-steel, extending 3 feet 6 inches above and 4 feet below the 23½-foot load line. The thickness amidships is 12 inches at the top for a depth of 4 feet 6 inches, and tapers thence to 8½ inches at the armor shelf. The maximum thickness of this side belt is maintained from the after barbette, where it begins, to the forward coal bunker bulkhead, tapering to 9 inches at about the center line of the forward barbette, thence to 4 inches in a length of 40 feet, which thickness is maintained to the bow. The diagonal armor, 9 inches thick, extends from the slopes of the protective deck to the top of the armor belt at the forward and after ends of the machinery spaces. Above the side belt is the casemate armor, 6 inches thick, worked from about the center line of the forward barbette to about the center line of the after barbette. The ends of this belt are formed by diagonal armor, 6 inches thick, worked from the sides of the vessel to the barbette armor. The superstructure armor is 6 inches thick.

The protective deck between the armor bulkheads consists of two courses of ½-inch steel plates, to which is riveted a $1\frac{9}{16}$-inch nickel-steel plate. At the ends of the ship this deck consists of one course of 1-inch steel and of a nickel-steel plate which is 2 inches thick forward and 3 inches thick aft; amidships, the nickel-steel is $1\frac{9}{16}$ inches thick, except over the steering engine, where it is 3 inches thick. The forward conning tower is 10, and the after one 6 inches thick.

The turrets for the 12-inch guns are of the balanced type, the armor for them and for the barbettes being of Harveyized nickel-steel, 12 inches thick.

ORDNANCE.

Main Battery.

4 12-inch breech-loading rifles.
16 6-inch rapid-fire guns.

Secondary Battery.

6 3-inch rapid-fire guns.
8 3-lb. rapid-fire guns.
2 1-lb. rapid-fire guns.
4 1-lb. automatic rapid-fire guns.
2 Colt machine guns.
2 3-inch field pieces.

Torpedoes.

2 submerged torpedo tubes.

MAIN ENGINES.

There are two four-cylinder triple-expansion engines of the vertical, inverted, direct-acting type, placed abreast of each other in separate watertight compartments, one L.P. cylinder being forward, followed by the H.P., I.P. and the second L.P. cylinders. Each cylinder is supported by four forged, nickel-steel, cylindrical columns, tied together by diagonal and fore-and-aft rods. The cast-iron guide for the crosshead is bolted between the outboard columns, these carrying also the brackets for the reversing shaft.

All valves are of the single-ported piston type, of cast iron, and made with one packing ring and follower. The H.P. and L.P. valves take steam from the outside, and the I.P. valves from the inside. They are arranged to draw out through the bottom head. The high-pressure cylinder has one, and each intermediate and low-pressure cylinder two valves. The valves are provided with balance pistons, the cylinders of which form part of the upper covers of the valve chests, the diameter being 6 inches for the H.P., and 8½ inches for the I.P. and L.P. balance pistons. The valve gear is of the Stephenson type, with double bar links. The H.P. valve stem takes hold of the link blocks direct, while the I.P. and L.P. valve stems are secured to a crosshead.

Independent adjustable cut-off blocks are fitted to each link, with a range of from .5 to about .77 of the stroke for the H.P., .73 for the I.P., and .7 for the L.P. cylinders. There are no starting valves on the cylinders, provision being made for start-

ing the engines by admitting live steam to the receivers. All cylinders are steam-jacketed around the working liners, and the I.P. and L.P. cylinders at both ends. Each engine has a 13-inch throttle valve, balanced by a piston working in a cylinder forming part of the throttle-valve casing, steam being admitted to the back of this balance piston. All steam pipes of and above 6 inches in diameter are of steel, smaller ones and all exhaust pipes are of copper.

The pistons are of cast steel and conical in shape, each I.P. and L.P. piston being fitted with a follower and two packing rings, 1½ inches wide and ⅝ inch thick, set out by steel springs. On the H.P. piston, which was at first fitted like that of the others, the two rings were changed, after trial, to one solid packing ring, 3 inches wide, with two circumferential grooves. The piston rods are hollow, of forged nickel-steel, oil tempered and annealed. They are 7 inches in outside diameter, the hole being 3 inches in diameter, except at the piston end, where it is 1¼ inches. The crosshead is forged on the piston rod, the cast-steel slipper being bolted to the head. The go-ahead guides, which are secured to crossbars on the engine columns at the bottom, and at the top to lugs cast on the cylinder casings, are of cast iron, and are hollow for the circulation of water. The backing guides are bolted to flanges on the go-ahead guides.

The connecting rods are of forged nickel-steel, oil-tempered and annealed, and have a 4½-inch hole bored throughout their length. The upper end is forked to span the crosshead, the rod being shrunk on the pin. The eccentrics are of cast iron, each in two parts. Each backing eccentric is securely keyed on the crank shaft, and each go-ahead eccentric is secured to its backing eccentric by two through bolts in a slotted hole, the hole being filled up after the eccentrics are set. The eccentricity is 4¾ inches. The eccentric straps are of composition, lined with white metal. The eccentric rods are of forged steel, the distance from centers of eccentrics to centers of link pins being 7 feet 10 inches. Each engine has a steam floating-lever reversing engine, which can also be worked by a hand pump through its hydraulic control cylinder.

The turning engine is a double inverted engine, which drives, by worm gearing, a second worm, which may be made to mesh at will with a worm wheel on the propelling shaft. The turning-engine shaft is squared at the end and fitted with a ratchet wrench for turning by hand.

The working platforms are inboard, communication between the engine rooms being possible through a vertical watertight door in the central bulkhead.

ENGINE DATA.

Cylinders, number for each engine	4
H.P., diameter, inches	34¾
I.P., diameter, inches	53
L.P., diameter, inches	63
Stroke of all pistons, inches	48
Valves, H.P. (one for each cylinder), diameter, inches	19½
I.P. (two for each cylinder), diameter, inches	19½
L.P. (two for each cylinder), diameter, inches	22
Valve stems, forged nickel-steel, diameter, inches	2⅞
diameter through valve, inches	1⅞
diameter at balance pistons, inches	1½
Main steam pipe, diameter at throttle, inches	13
area of cross section, square inches	132.73
Volume swept by H.P. piston, mean, per stroke, cubic feet	25.81
I.P. piston, mean, per stroke, cubic feet	60.75
both L.P. pistons, mean, per stroke, cubic feet	172.11
Net area of I.P. to H.P. piston	2.354 to 1
both L.P. to H.P. pistons	6.668 to 1
Clearance of H.P. cylinder, per cent	top, 20.17; bottom, 22.35
I.P. cylinder, per cent	top, 19.63; bottom, 23.92
F.L.P. cylinder, per cent	top, 15.13; bottom, 18.33
A.L.P. cylinder, per cent	top, 14.82; bottom, 18.44
Piston rods, diameter, inches	7
of axial holes, inches	3
length, feet and inches	8– 9⅝
Cylinder walls of H.P., I.P. and L.P. cylinders, thickness, inches	1½, 1⅝, 1¼
liners of all cylinders, thickness, inches	1¼
Valve-chest liners, thickness, inches	1 1/16
Jacket space, inch	⅝
Cylinder relief valves (two on each cylinder), diameter, inches	3½
Connecting rod, length, center to center, inches	96
diameter, upper end, inches	7¼
lower end, inches	8¼
axial hole, inches	4½

Connecting rod, crosshead bolts (2), diameter, inches	4¼
crank pin bolts (2), diameter, inches	4¼
Crosshead, surface, ahead, square inches	540
backing, square inches	432
pin, diameter, inches	10½
diameter of axial hole, inches	5½
length of pin, inches	16
Crank shaft, number of sections	4
diameter, inches	15½
of axial holes, inches	8
coupling discs, inches	28
thickness coupling discs, inches	3½
coupling bolts in one flange (12), diameter, inches	3
journals, diameter, inches	15½
length, inches	17
length of H.P. and I.P. sections, feet and inches	8– 1½
length of each L.P. section, feet and inches	8– 8½
maximum distance beyond journals, feet and inches	3– 5¼
pin, diameter, inches	16
length, inches	18
projected area, square inches	228
diameter, axial holes, inches	8
webs, width, inches	18
thickness, inches	10
Thrust shaft, diameter, inches	15
axial hole, inches	8½
collars, number on each shaft	15
diameter, inches	23
thickness, inches	2
space between, inches	3½
total surface both engines, square inches	7,162.86
length, feet and inches	13– 6
steady bearings, inches	14
Line shaft, diameter, inches	15
axial hole, inches	8½
coupling discs, inches	29
coupling bolts in one flange (8), diameter, inches	3½
length, feet and inches	24– 9½
Propeller shaft, diameter in strut bearing, inches	15½
after stern-tube bearing, inches	15 9/16
forward stem-tube bearing, inches	15½
length, feet and inches	59– 1
Inboard collar coupling, diameter, inches	29
length, inches	21½
of collar, inches	15
Stern bracket bearing, diameter, inches	17½

Stern bracket bearing, length, feet and inches	4- 3
Reversing engine, diameter of steam cylinder, inches	15
controlling cylinder, inches	7½
stroke, inches	20
Turning engine, diameter of steam cylinders (2), inches	7
stroke of pistons, inches	5

Shafting and Bearings.—The crank shafts are of forged nickel-steel, oil tempered and annealed, in four sections, with cranks at 90 degrees, the sequence for the ahead motion being H.P., A.L.P., F.L.P., I.P. The thrust, line and propeller shafts consist of one section each for each engine. There are two spring bearings for each line shaft.

The thrust bearings are of the horseshoe pattern, the ends and sides of the cast-iron pedestals forming an oil trough, with a steady bearing lined with white metal at each end to take the weight of the shaft. The horseshoes are of cast steel, lined with white metal, properly channeled for oil, and are adjusted by brass nuts on steel side rods, 3¾ inches in diameter. An oil cup is fitted on top of each horseshoe, with an oil hole to each collar. At each end of each thrust bearing is a divided stuffing box and gland of composition to prevent the escape of oil. The pedestal is bolted to a cast-iron sole plate, fitted with forged-steel wedges at each end of the pedestal for adjusting the bearing fore-and-aft.

The forward part of the propeller shafts, within the stern tubes, and the after part of these shafts, from about 6¼ inches forward of the stern bracket bearing to 1 inch inside of propeller hub, are covered with a composition watertight casing, shrunk and pinned on. The inboard coupling of each propeller shaft consists of a forged-steel collar, secured to the shaft by four steel keys. Abaft of this is another collar, in halves, which is secured to the intermediate collar and to the coupling on the line shaft by eight fitted steel bolts.

Main Condensers.—There is one main condenser in each engine room. The shell is of steel, $\frac{5}{16}$ inch thick, stiffened with angle bars, butted with double butt straps, and with an angle iron riveted around each end to form flanges for securing the water chests. The latter are of composition, and are lined with rolled

zinc plates. The water chest at the injection and delivery end has a division plate fitted with a valve, which is worked by a lever on the outside of the condenser, to allow the circulating water to pass directly overboard when this valve is open. Three steel baffle or deflecting plates direct the exhaust steam over all of the tubes. The latter are supported in the middle of their length by a steel plate, the holes in which are fitted with Muntz metal thimbles. The tubes are of seamless-drawn, and the tube sheets of rolled Muntz, metal. The tubes are tinned inside and outside, and are packed with brass screw glands and cotton tape. To the top of each condenser is riveted a composition nozzle, which contains two openings for the main exhaust, each 22 inches in diameter. On the forward side of the nozzle there is a 3-inch opening for the main air-pump exhaust, and one of 5 inches for the bleeder pipe. The after side has a 9-inch opening for the auxiliary exhaust, one of 4 inches for the main circulating-pump exhaust, and one of 2 inches for soda and L.P. balance-piston connections. On the upper side of the nozzle, in the middle, there is a 3-inch opening for a spring safety valve, the latter being loaded to 25 pounds above the atmosphere. There is a 12-inch opening in the bottom of each condenser for the suction pipe to the main air pump, and a 12-inch manhole. In the plate of the latter there is a 2-inch hole for a connection from the auxiliary steam pipe for boiling water in the condenser when cleaning tubes. There is a 11 by $5\frac{1}{2}$-inch hand hole in the top, and a 7 by $3\frac{3}{4}$-inch hand hole in the bottom of each tube sheet. Drain cocks are fitted to heads and shell.

Diameter of shell (inside), feet and inches	6– 5
Thickness of shell, inch	$\frac{5}{16}$
Length over heads, feet and inches	14– 5
Tubes, diameter (outside), inch	$\frac{3}{4}$
length between tube sheets, feet and inches	12– 3
thickness, No. 18, B.W.G., inch	.048
spacing between centers, inch	$1\frac{1}{16}$
number in each condenser	4,680
Condensers (2), cooling surface, square feet	18,762

Main Air Pumps.— For each main engine there is a Blake cross-compound, vertical, double-acting air pump. The piping

and valves are so arranged that, by means of a three-way cock, the pumps can be run compound or simple. In the latter case, either side alone can be run as a single pump, or both sides can be run together as two single pumps. There are no suction valves, and the inlet port for the water is on the middle of the length of the pump cylinders. The plunger has no valves, and is 16¼ inches long. The upper working lining of the pump cylinder is 16⅞, and the lower one 16¾ inches long, a 2-inch space being left between them.

A copper suction pipe, 12 inches in diameter, with a straight-way valve near the pump, connects each air pump to its condenser, and a 10-inch delivery pipe leads to the corresponding filter tank. The air-pump casings are of composition, the steam cylinders of cast iron. The H.P. piston is packed with two snap rings, and the L.P. with two rings set out by flat springs. The pump valves are composed of three thin, flat discs of rolled manganese-bronze held in place by a guard and spiral springs, the lift being ⅜ inches.

Diameter of H.P. cylinder, inches	12
L.P. cylinder, inches	18
pump cylinders (2), inches	22
Stroke, inches	18
Diameter of piston rods (nickel-steel), inches	2 9/32
pump rods (Tobin bronze), inches	2¼
plunger, inches	22
pump valves, inches	3½, 3¾, 4
Number of delivery valves in each end	15
Area through 15 valve openings, square inches	6.42
by lift, square inches	4.05

Main Circulating Pumps.—For each condenser there is a centrifugal, double-inlet, circulating pump, driven by two simple engines with cranks at 180 degrees to each other. The valve is of the piston type, taking steam on the inside. Each pump is capable of discharging 12,000 gallons of water per minute from the bilge when the engines are making about 250 revolutions per minute. Each pump draws water from the sea, the main drain or the engine-room bilge, and discharges into the condenser or directly overboard through the valve in the division plate in the

water chest at the injection end of the condenser. There are stop valves in the suction pipes leading from the sea, the main drain and the bilge in each engine compartment. These valves are so connected by a locking device that when one is open the others must be shut.

Diameter of steam cylinders (2) inches	10
Stroke, inches	10
Diameter of pump runner, inches	48
Width of pump runner at periphery, inches	3¼
hub, inches	9
Diameter of inlet and outlet nozzles, inches	17

Feed and Filter Tanks.—There is in each engine room, under the main condenser, a feed tank of 2,500 gallons' capacity, a filter tank of 1,020 gallons' capacity being combined with it, the feed water, after passing through the filter, flowing to the feed tanks. These tanks are of ¼-inch galvanized steel. The feed tank is fitted with zinc protectors, the usual covers, glass water gauge and shut-off and drain cocks, and is well braced internally. It has pipe connections for an overflow pipe to bilge, with spring-loaded valve; a suction pipe to feed pumps, drain pipes from traps, and a 3-inch copper vapor pipe, with its discharge above the level of the awnings. A balanced valve, operated by a copper float or by a lever from the outside, is fitted to the suction end of each feed tank, and is so arranged that no air can enter the feed pipes.

The filter tank is divided by partition plates into three chambers containing the filtering material, which, in this case, is Loofa. A 10-inch discharge pipe from the main and a 5-inch discharge pipe from the auxiliary air pumps lead into the filter tank in the same engine room. Each filter chamber has a 2-inch vapor pipe, these continuing into the 3-inch vapor pipe from the feed tank. Drain cocks are provided for each chamber.

Feed Pumps.—In each engine room there is a main feed pump capable of delivering 640 gallons per minute, and in each of the four boiler compartments an auxiliary feed pump of 400 gallons capacity per minute. The main feed pumps take steam from both the main and auxiliary steam pipes, and draw water from

the feed tanks, discharging it through the feed-water heaters into the main feed pipes. The auxiliary feed pumps draw from the feed tanks, the sea, the drainage system, the fire-room bilges or the boilers, and discharge into the boilers, fire main, or overboard through sea valves in their own compartments. All feed pumps are of the Blake, simplex, vertical, piston type. The water valves are metalic.

Main feed pumps (2):	
Diameter of steam cylinders, inches	16
water cylinders, inches	12⅝
piston rods, inches	2¾
Stroke, inches	18
Auxiliary feed pumps (4):	
Diameter of steam cylinders, inches	14
water cylinders, inches	10
piston rods, inches	2½
Stroke, inches	12

Feed-water Heaters.—There is in each engine room a feed heater, containing 804 square feet of heating surface. There are 602 ⅝-inch composition tubes, not tinned, in each heater, the tubes being staggered and expanded into both tube sheets. The shell and heads of the heater are of composition, and the seven deflecting plates of rolled brass. Steam is taken from the auxiliary exhaust line and passes through the tubes; there is a relief valve to the main condenser. The main feed pumps discharge through the heaters, a by-pass cutting out the heater when desirable.

Pumps in the Engine Rooms.—In each engine room there are two pumps, each of 400 gallons per minute capacity, to be used as fire pumps with steam of eighty pounds pressure. One of these pumps is to draw water from the bilge or the sea and to discharge into the fire main or overboard direct; the other draws from the sea only and may discharge into the engine-room water-service pipes, the fire main, or the distiller circulating pipes, through which it may be passed into the flushing system or overboard.

Number and type of pumps	4, Blake, vertical, simplex.
Diameter of steam cylinders, inches	14
water cylinders, inches	10
piston rods, inches	2¼
Stroke, inches	12

There is also in each engine room a hotwell pump of about 640 gallons per minute capacity, of light-service proportions, which draws water from the feed tank and discharges it into the feed-pump suction pipe in the same engine room. In addition, these pumps draw from the air-pump channel ways, from the sides of the ship above the water line for taking in fresh water, and from the reserve tanks. Discharge pipes are fitted also to the reserve tanks.

Number and type of pumps	2, Blake, vertical, simplex.
Diameter of steam cylinders, inches	12
water cylinders, inches	14
piston rods, inches	2
Stroke, inches	12

On each main engine shaft there is one sanitary and one bilge pump, each worked by an eccentric. The bilge pumps draw from the engine-room bilges and crank pits, discharging overboard. The sanitary pumps draw from the sea and deliver into the distiller circulating pipes, overboard, or into the flushing system.

Auxiliary Condensers.—In each engine room, inboard on the upper grating, there is a Wheeler auxiliary condenser, with combined horizontal double-acting air and circulating pump. Shell, heads and tube plates are of composition. The packing, diameter and spacing of tubes are the same as in the main condensers. The air pump discharges into the feed and filter tank.

Cooling surface, both, square feet	2,002.5
Diameter of steam cylinder of pump, inches	14
circulating-pump cylinder, inches	12
air-pump cylinder, inches	12
Stroke, common, inches	12
Air pump, kind of valves	rubber

Screw Propellers.—The propellers are of manganese-bronze, the starboard propeller being right and the port one left-handed.

They are true screws, accurately balanced, with pitch adjustable from 16¾ to 18¾ feet. Each boss is accurately bored to fit the taper on the end of the shaft and fitted with two feather keys, being held on the shaft by a nut screwed on and locked in place. Each boss is fitted at the after end with a composition cap bolted on watertight; bosses and caps are finished all over, tinned and smoothed. The blades are first smoothed and polished, then tinned, and next again smoothed.

Number of blades	3
Diameter, feet	17–
of hub, inches	52½
Length of hub, inches	40½
Pitch, on builder's trial, feet and inches	17– 9
as set on official trial, mean, feet and inches	18– 2.21
Helicoidal area, square feet	73.7
Disc area, square feet	216.98
Diameter÷pitch	1.07

Boilers.—There are twelve Thornycroft boilers of the "Ohio" type arranged in four compartments, as shown in the plan, giving four large and four small fire rooms and three smokepipes. The boilers and connections were designed and built to carry a pressure of 250 pounds per square inch, the working pressure at the engines being lowered to 200 pounds by means of reducing valves. After the boilers and pipes were connected in the vessel, it was decided, as a result of the steam test, to take out the reducing valves, lower the working pressure in the boilers to 225 pounds, and run with a pressure of about 210 pounds at the engines.

The general construction of the pressure parts of the boiler is shown in the accompanying plate, the casing being removed On top of each boiler there is an air heater, which is open to the fire room at the front end, and connected, at the back, to an air duct, which is built on the boiler casing and leads to the back of the ash pit. The closed fire-room system of forced draft is used, the air blown into the fire rooms being forced through the heaters and down the duct at the back to each ash pit. Two dampers are fitted at the bottom of each air duct, and the ash-pit doors are kept closed.

THORNYCROFT BOILER.

The steam and water drums are of open-hearth steel with welded joints. The tubes are of seamless cold-drawn steel, expanded at both ends into radial holes in the drums. The two rows of tubes on each side, which form the furnace and combustion chamber walls, are larger in diameter and heavier than the others. At the top of each combustion chamber a brick baffle, 9 inches deep, is fitted into the triangular space below the meeting point of the fire-tube walls. The two rows of tubes forming the outside wall on each side of the boiler rise above the highest point of the inside of the steam drum. The top of the inside of the outer row is about $3\frac{3}{8}$ inches, and of the next row about $\frac{1}{2}$ inch above the water in the steam drum, when the latter is quite full, thus preventing the filling of these tubes. This defect is overcome in the later Thornycroft-Schulz type. The steam drums are fitted with the Thornycroft feed-water regulator and the usual steam and water gauges.

The grate bars of cast iron, 2 feet $11\frac{1}{2}$ inches long, made double and single, with flat tops $\frac{1}{2}$ inch wide. The air space between bars is $\frac{1}{2}$ inch at the top and $\frac{3}{4}$ inch at the bottom. Bearer bars are of wrought iron.

The furnace fronts are made with double walls of steel plates and angles, with three fire and two ash-pit door frames for each half of each boiler. The space between the two walls communicates with the ash pits, and the upper part of each furnace-door frame is perforated for supply of air above the grates. Two fire extinguishers are fitted to each furnace.

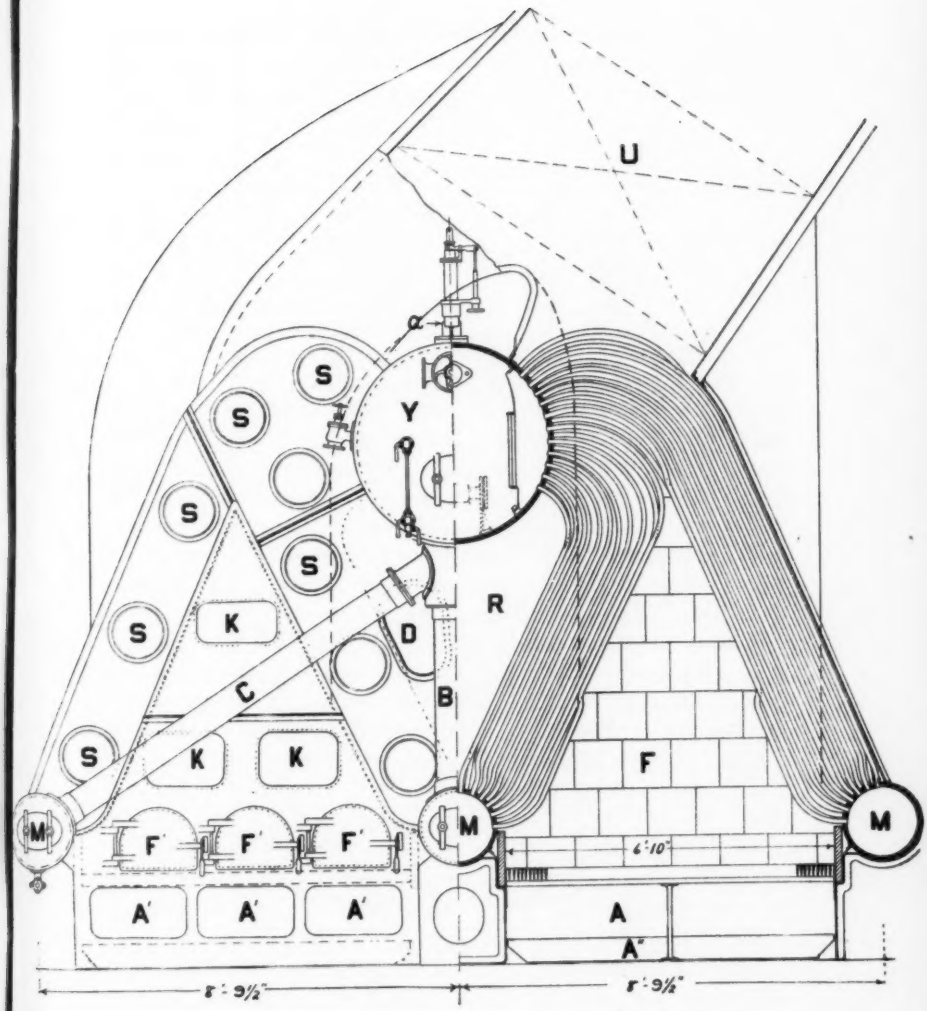
Owing to the contracted space allowed for the boilers, there is no passage at the fire-room level between the boiler compartments. A narrow platform, 5 feet above the fire-room floor, and reached by little ladders, runs above the outboard water drums of two adjoining boilers. There is no passage on the inboard side of the boilers.

Like in most modern ships, there are no fire-room *ladders*, iron rounds riveted to the vertical bulkhead furnishing the only direct means of going from the fire room to the upper gratings and fire-room hatches. These rounds are fitted "to afford a ready escape for firemen and coal passers," as the official speci-

fication has it; but, as the hand rails become so hot that they cannot be touched without asbestos gloves, these "ladders" are practically of no use during steaming. These hand rails should be made of piping, through which a flow of water could be maintained, so that the "ladders" could be used at all times.

Number of boilers	12
furnaces	24
Steam drum, inside diameter, inches	48
thickness, inch	1
Water drums, inside diameter, inches	19
thickness, inch	$\frac{3}{4}$
Number of fire-wall tubes, one boiler	220
other tubes in one boiler	1,184
Fire-wall tubes, O.D., inches	$1\frac{5}{8}$
gauge, B.W.G.	No. 9
All other tubes, O. D., inches	$1\frac{1}{4}$
gauge, B.W.G.	No. 10
Grates, length, feet	6
width, feet and inches	6- 9
distance of top from center of water drums, inches	10
Effective heating surface, one boiler, square feet	4,281
grate surface, one boiler, square feet	81
Total heating surface, square feet	51,372
grate surface, square feet	972
Ratio of heating surface to grate surface	52.85
Working pressure, pounds per square inch	225
Safety valves (24), diameter, inches	3
Sentinel valves, diameter, inch	$\frac{1}{2}$
blowing-off pressure, pounds	254
Smoke pipes (3), diameter, feet and inches	7-10$\frac{5}{16}$
area, each, square feet	48.51
Total, square feet	145.53
height above grates, feet and inches	94- 4
Escape pipes (3), diameter, inches	7

Forced Draft.—In each double fire room there are two Sturtevant blowers and one in each single fire room, designed for five hundred revolutions per minute. They deliver into the fire rooms as explained above. The blowers are placed in front of the boilers, immediately under the protective deck, and therefore in about the hottest place in the fire room. In the armored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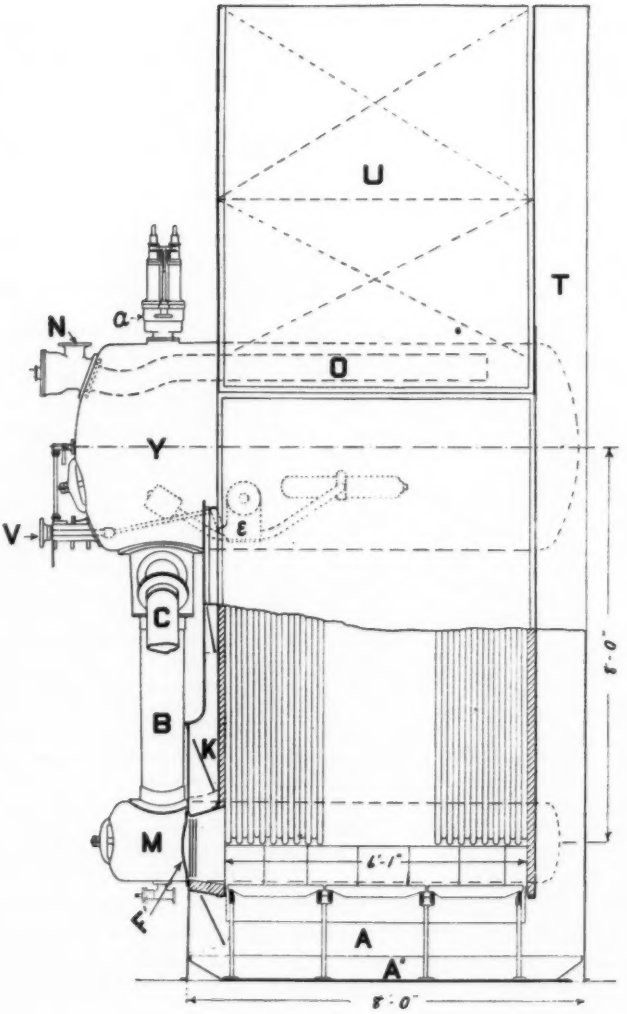
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8'-0"

cruisers building at Newport News a great improvement has been made in locating these blowers above the protective deck, where they are readily accessible and can be taken care of while running.

Number of blowers	12
Type of blower engines	vertical, double enclosed.
Steam cylinders (2), diameter, inches	5
Stroke, inches	4
Diameter of piston rods, inches	1½
fan, inches	60
Area of induction nozzle, square inches	1,060
eduction nozzle, square inches	1,740

Steam Traps.—The separators, jackets, feed-water heaters, main and auxiliary steam pipes, the radiators and all places where condensed steam can accumulate are fitted with drain pipes and cocks, or valves, with Kieley automatic traps which discharge into feed tanks. The traps have the usual by-pass pipes and valves for convenience in overhauling.

Engineer's Workshop.—The following tools are provided: One 30-inch screw-cutting extension lathe, 8 feet between centers when gap is open; a 14-inch screw-cutting lathe, 40 inches between centers; a column-shaping machine of 16 inches stroke and 19½ inches traverse; a double-geared drilling machine, with screw feed; a small iron-column bench drilling machine for hand and power; a combined hand punch and shears, and an emery grinder, with two wheels, 12 inches in diameter and 2-inch face. These tools are driven by a vertical engine with fly-wheel, driving pulley and automatic governor.

Ash Hoists.—The ventilator in each fire room is fitted with vertical guide strips on the inside for hoisting ashes. The ash-hoisting engines, Williamson Bros.', are on the berth deck in the fire-room hatches, and can hoist 300 pounds from the fire-room floor to the main deck in five seconds, with steam of 80 pounds pressure. The engines have two cylinders, 4½ by 4½, with cranks at right angles; the reversing gears are worked from the deck, and have safety gears to prevent overwinding, and to stop when the bucket reaches the fire-room floor.

Distilling Apparatus.—This consists of four evaporators and two large distillers, with a small supplementary distiller, having a combined capacity of 10,000 gallons of potable water per twenty-four hours at a temperature of not more than 90 degrees Fahrenheit. The evaporators take steam from the auxiliary steam pipe. The heating surface of each evaporator is 260.3 square feet; the cooling surface of each large distiller is 123.4, and of the supplementary distiller, 22.3 square feet.

There are two feed pumps for the evaporators in the same room, of the Blake, vertical, simplex type, with steam cylinder 4½ and water cylinder 4 inches in diameter, and a stroke of 6 inches. There are two Blake, horizontal, combined fresh-water and brine pumps in the evaporator room, with steam cylinder 4½ and water cylinders 3 inches in diameter, the common stroke being 4 inches. The fresh-water pump draws from a small reservoir tank and discharges into either the fresh-water or main feed tanks.

Ice Machine—There is one Allen dense-air ice machine, capable of producing the cooling effect of two tons of ice per day, which is connected to the ice tank, the very large cold-storage rooms and the scuttle butt.

Cylinders, steam, diameter, inches	9¼
compressor, diameter, inches	7
expander, diameter, inches	5¼
Stroke, inches	13

Ventilation.—Two blowers, operated by electricity, are fitted in a separate room below the protective deck and just forward of the forward engine-room bulkhead to force air into the engine rooms. These have a capacity of about 12,000 cubic feet per minute. To ventilate the ship generally, six blowers, each having a capacity of about 8,000 cubic feet per minute, arranged in independent groups of two and operated by electricity, are provided. These groups are used on the plenum system under ordinary conditions of service. In action, the forward group may be used to force air into the forward magazines and ammunition rooms, the after group into the after magazines and handling rooms, and

the midship group into the ammunition passages at the sides of the vessel. Special electric blowers and pipes ventilate the dynamo room, torpedo room, steam-steering room, engineers' workshop, etc.

Windlass, Winches, etc.—There is a vertical, Hyde steam windlass, situated forward between the main decks, and a steam winch in the same house. The windlass engine has two cylinders, 15 inches in diameter, the stroke being 14 inches. There is a steam winch on the main deck aft and two on the upper deck aft. All winches have two cylinders, 8 inches in diameter, with a stroke of 8 inches. The ship is provided with four electric boat cranes.

Steering Engine.—On the after platform deck there is a Williamson Bros.' combined hand- and steam-steering engine; diameter of cylinder, 15 inches; stroke, 12 inches. The engine is controlled from the conning tower and pilot house by means of wire rope.

Steam Cutters.—There are two steam cutters, one 40 and the other 36 feet long. They have Ward boilers, compound engines and keel condensers.

No. of cutter.	Length.	No. of engine.	Size of engine.	Boiler, No. and type.	G.S.	H.S.	Weights.	
	feet.		*inches.*		*sq. ft.*	*sq. ft.*	*pounds.* Eng.	Blr.
487	40	No. 424 E_2	$\frac{5 \times 10}{8}$	No. 1245 G_1, round.	9.37	207	950	2,300
453	36	No. 469 G_2	$\frac{4 \times 8}{6}$	No. 1100 G_1, square.	9.00	182	420	2,778

Torpedo Compressors.—There are two Rand type VI air compressors for torpedoes, driven by electricity. Each motor is shunt wound for an 80-volt current to develop 25 horsepower on 300 revolutions per minute. The compressor can deliver 20 cubic feet per hour at a pressure of 1,700 pounds per square inch.

Electric Plant.—The installation consists in general of four 50-kilowatt and four 32-kilowatt generating sets, all of 80 volts pressure at the terminals; 770 incandescent lamps, four 30-inch searchlights, two sets of electric night-signaling apparatus, two

diving lamps, four portable electric ventilating sets, twelve $\frac{1}{12}$-horsepower electric desk or ceiling fans, with the necessary switch boards, wire, wiring accessories, molding, conduit tubes, fixtures and lamps, instruments, tools, spare parts and stores usually furnished for the proper manipulation, test and repair of the plant. The generating sets are secured to a common bedplate, and engines and dynamos directly connected. The dynamos are of the direct-current, compound-wound, multipolar type, and made by the General Electric Co. The wiring is on the 2-wire feed system.

DYNAMO ENGINES.

Type	Compound Tandem, L.P. cylinder on top.
32-K.W. dynamo engines, No.	4
Diameter of cylinder, H.P., inches	9
L.P., inches	15
Stroke, inches	6
Diameter of piston rods, H.P., inches	$1\frac{5}{8}$
L.P., inches	$1\frac{7}{8}$
50-K.W. dynamo engines, No.	4
Diameter of cylinders, H.P., inches	$10\frac{1}{2}$
L.P., inches	18
Stroke, inches	8
Diameter of piston rods, H.P., inches	2
L.P., inches	$2\frac{1}{8}$

SPEED TRIAL.

The vessel left Newport News, Va., on Saturday, October 17, 1903, for Boston, Mass., but was delayed about one day at Old Point by weather conditions; 474 miles of the run up were made at an average speed of 14.96 knots, the mean average revolutions being 92.42 per minute. Forced draft was put on for about $1\frac{1}{4}$ hours to drill the crew.

The official speed trial took place on the usual course off the New England coast, near Cape Ann, on October 21. The course was 33 nautical miles. As the double run, including the turn, was made in less than four hours, the full-power trial conditions were kept up long enough at the end of the down run to make

up the required contract time. Weather conditions were satisfactory, except that there was a strong breeze on the bow during the last part of the run up. The machinery worked satisfactorily. The S.I.P. crank pin and crosshead brasses heated, the other journals running cool, and there seemed to be no necessity for more than ordinary lubrication. On this point of lubrication during trial trips, it is pertinent to ask if the usual method is necessary. Aside from the dirty condition of both men and engine rooms, the reckless use of lubricants, and often water, makes the engine rooms dangerous for the attendants and does not add to the efficiency of the lubrication. At high speed, oil will, of course, be thrown off from the crank pins, but with properly-fitted guards, which still leave the journals open to test by hand, and a generous but not reckless supply of clean lubricant, better lubrication could be assured and the danger and disagreeable features of the usual method eliminated. Then, should water have to be turned on the running parts at any time, the results of its use, so far as floor plates and platforms and the oilers are concerned, could easily be removed.

There were put in, for use during the trial, about twenty-five extra electric lights in each engine room.

The absence of vibration during the trial was very marked.

The force employed in the engine department for the trial consisted of 246 men, or 34 more than the working complement of the engineer force when the vessel is commissioned.

The firing was good, but not rapid enough during part of the trial. At no time was the steam pressure up to the limit for more than a few moments. Owing to a long delay, due to the fouling of the anchor while steaming for the first stake boat, the fires were not in as good condition as they would and should have been. The allowed air pressure of 2½ inches of water was used only on the two last laps of the run down, and did not exceed 2.1 inches on the run up.

	Run up.	*Run down.*
Elapsed time, hours, minutes and seconds	1·52·15.5	1·47·9
hours	1·871	1·786
Total revolutions, S. engine	13,160	12,430
P. engine	13,204	12,539

	Run up.	*Run down.*
Total revolutions, mean of both engines	13,182	12,484.5
Average revolutions per minute, both engines	117.42	116.52
Mean speed, knots, uncorrected	18.05	
with tidal corrections	18.1454	
Draught of vessel, mean, during trial, ft. and ins	23-6	
Displacement corresponding to this draught, tons	12,240	

PERFORMANCE—SYNOPSIS OF STEAM LOG.

Performance :	
Steam pressure at boilers (per gauge), pounds	208
H.P. steam chest, P. engine (per gauge), pounds	186.83
1st receiver, S. engine, absolute, pounds	98
P. engine, absolute, pounds	95.3
2d receiver, S. engine, absolute, pounds	33.9
P. engine, absolute, pounds	33.1
Vacuum in condensers, in inches of mercury	P., 25.05 ; S., 26.22
Main effective pressures in cylinders in pounds, per square inch :	
Main engines—starboard, H.P.	79.25
I.P.	49.63
forward L.P.	16.64
after L.P.	17
port, H.P.	79.81
1st I.P.	45
forward L.P.	14.9
after L.P.	15.15
Mean effective pressure, in pounds per square inch, on L.P. piston, equivalent to aggregate M.E.P. on all pistons, starboard	46.18
Mean pressure, in pounds per square inch, on L.P. piston, equivalent to aggregate M.E.P. on all pistons, port	42.88
Auxiliary machinery, mean effective pressures :	
Pumps—air, main, H.P.	14.78
L.P.	11.46
circulating, main	port, 47.38 ; starboard, 49.94
feed, main	117.5
air, auxiliary	16.54
Dynamo engines, 32 kw. ; H.P., 29.44 ; L.P., 11.63. ; 50 kw. ; H.P., 40 ; L.P.,	10.063
Blower engines	43.56
Revolutions per minute :	
Main engines	starboard, 116.61 ; port, 117.32
Pumps, air	starboard, 27.25 ; port, 27
circulating	starboard, 131 ; port, 128.5
feed, double strokes per minute	23.67
water service, double strokes per minute	31.25
sanitary, double strokes per minute	23

Blower engines	539
Speed of ship in knots per hour	18.1454
Slip of propeller in per cent. of its own speed, based on mean pitch.....port, 13.288; starboard,	13.313
Air pressure in fire room, in inches of water..	2.2
Indicated horsepower:	
Main engines, starboard, H.P.	2,072.9
I.P.	3,067.6
forward L.P.	1,457.2
after L.P.	1,488.8
total	8,086.5
port, H.P.	2,109.5
I.P.	2,798.6
forward L.P.	1,312.7
after L.P.	1,334.8
total	7,555.6
Air pump.....starboard engine, 12.863; port engine,	12.975
Circulating pump.....starboard engine, 49.6; port engine,	47.04
Feed pumps (2 main)	81.26
Hotwell pumps (2)	14.94
Auxiliary condenser, air and circulating pumps, etc.	5.62
Water-service pumps (2)	9.18
Blower engines for forced draft in fire rooms, 12 at 22.593 =	271.12
Dynamo engines, 1 — 32 kw. = 51.997; 1 — 50 kw. = 77.976	129.974
Collective, of both main engines	15,642.1
main engines, air, circulating and feed pumps and auxiliary engines in operation during trial	16,276.67
all machinery during trial, per square foot of G. S.	16.743
H. S.	.3168
Main engines, air, circulating and feed pumps, per square foot of G. S.	16.303
Main engines, air, circulating and feed pumps, per square foot of H. S.	.3084
Coal—kind and quality.....New River, hand-picked,	excellent.
pounds per hour	38,248
I.H.P., collective, all machinery in operation..	2.35
I.H.P., collective, main engines, air, circulating and feed pumps	2.41
hour per square foot of G. S.	39.35
H. S.	.745
Cooling surface, square feet per I.H.P., main condensers only...	1.1558
and auxiliary condensers	1.278
Heating surface, square feet per I.H.P., collective, all machinery..	3.1563

After the completion of the trial and the return of the vessel to Newport News, the machinery was opened for inspection and found to be in excellent condition on the whole. A few furnace-door casings were found warped. The boilers were clean and in good condition.

The steering engine worked smoothly, putting the helm from hard-a-port to hard-a-starboard in 20 seconds.

THE DEVELOPMENT OF THE SUBMARINE IN THE DIRECTION OF INCREASED SCOPE.

A STUDY OF THE LAKE SUBMARINE TORPEDO BOAT *PROTECTOR*.

By Lieutenant John Halligan, Jr., Member.

THE MILITARY STRENGTH OF THE SUBMARINE.

The submarine, *when within torpedo range*, is the superior of the battleship, since the battleship is vulnerable to the torpedo—the weapon of the submarine. In the ability of the submarine to submerge itself to considerable depth, and thus secure the protection of an incompressible liquid medium, the craft possesses the power to assume an indefinite thickness of armor. This, combined with its invisibility, makes the submarine doubly secure from the gun—the weapon of the battleship.

The present state of development of the submarine, with its automobile torpedo, is such that when the boat is submerged and within torpedo range, it must be admitted that the disablement or destruction of the battleship is probable—the degree of probability and the amount of damage depending principally on the number of torpedoes that can be discharged by the submarine, and on her facilities for locating and estimating the range of the target. Thus the submarine, within the limitations given above, is eminently successful.

THE MEASURE OF ITS EFFICIENCY.

The value of the submarine as a formidable weapon of war will depend primarily upon three factors: First. The ability of the commander of the boat to place his vessel within torpedo range of the enemy. Second. The ability successfully to discharge the torpedo. Third. The character of the personnel.

In the present phase of submarine-boat development, considering the varying conditions of distance, state of sea and weather, as well as the probable speed and course of a watchful enemy, the power of placing the submarine well within torpedo range constitutes by far the most important factor of the problem. Inasmuch as the different types may be supposed to be armed with the same efficient and trustworthy type of torpedo, and as the daring and skill of the personnel may be equal for all types, it is seen that the true measure of the efficiency of any submarine boat is the ability successfully to place itself within torpedo range of the enemy. When the earnest call for the submarine comes, the final measure of comparative worth of existing types will rest upon their ability to attain a position of advantage against the enemy.

And the logical development of any type of submarine will be such as will tend to render the attainment of this position the more certain.

THE ATTAINMENT OF A POSITION WITHIN TORPEDO RANGE.

This involves—

(a) Ability of the boat to submerge quickly and at will.

(b) Absolute control of depth of submergence.

These are the properties which may be regarded as the fundamental desiderata of a submarine, and, therefore, in determining the comparative worth for naval purposes of distinct types of such construction the greatest values should be ascribed to the factors or qualities which in all vessels of war tend—

I. To increase the craft's scope or radius of action from a fixed base.

II. The ability to readily and rapidly change base.

III. The ability to abandon the base or to intercept the vessels of an enemy.

These qualities of efficiency and endurance in war vessels have likewise been thus expressed:

(a) Seaworthiness. In case of submarines, safety and stability in surface, awash and submerged condition.

(b) Speed. For submarines, surface and submerged.

(c) Cruising radius and habitability. The one useless without the other.

The navigational facilities. Particularly in submarines should this factor be regarded as of importance, since heretofore, in many types of craft, it has been maintained that the boat is blind.

SUBMERGENCE BY DIVING.

The first qualities demanded of the submarine are, naturally, those of ready submergence and ability to maneuver at will while submerged. Indeed, in most types of the past, this, except for ability to place or discharge a torpedo, was the only accomplishment hoped for, speed, seaworthiness and cruising ability being considered as foreign to the type.

Of the four methods of submergence developed in the history of submarines—namely, submergence by destruction of buoyancy, by vertical propellers, change of volume of displacement, diving, with slight reserve of buoyancy—the last survived.

It has been claimed by advocates of this method (and justly, so long as there was no proof to the contrary) that this furnishes greatest facility in submerging, and the best control of depth of submergence while underway.

THE SCOPE OF SUBMARINE OF THE PAST.

Heretofore it has been generally accepted (in this country at least) that the submarine, like the devil fish, must, to a great extent, lie in wait for its prey. In accordance with such estimate practically all official tests of submarines have been conducted in land-locked harbors, whose depth of water is less than that off shore, where it is presumed that the submarine would be expected to operate in time of war. The submarine has thus been granted the security of a harbor for base of operation, and its field of usefulness has been narrowed to the defense of that particular harbor. Changes of base as well as cruises to distant points have generally been made heretofore from the end of a towline.

AN ATTEMPTED ENLARGEMENT OF SCOPE.

The development of the *Protector* is of particular interest to the service in that she is an attempt at enlarging the scope of the submarine, principally along the headings outlined above.

The first and most difficult problem in the designing of such a boat was the combination of facility of submergence with seaworthiness under severe conditions. A sacrifice of longitudinal stability is essential in the diving type of submarines to render diving effective at the submerged speeds attainable. This instability must impair the sea-going qualities of the submarine, and it was to overcome this inherent disadvantage and weakness that the designer of the *Protector* effected a modification of means of submergence whereby the stability of the submarine would be increased and her radius of action extended.

The combination of facility of submergence with comparatively great longitudinal stability secured in the *Protector*, is the most remarkable feature of this interesting boat.

THE *PROTECTOR'S* PEDIGREE.

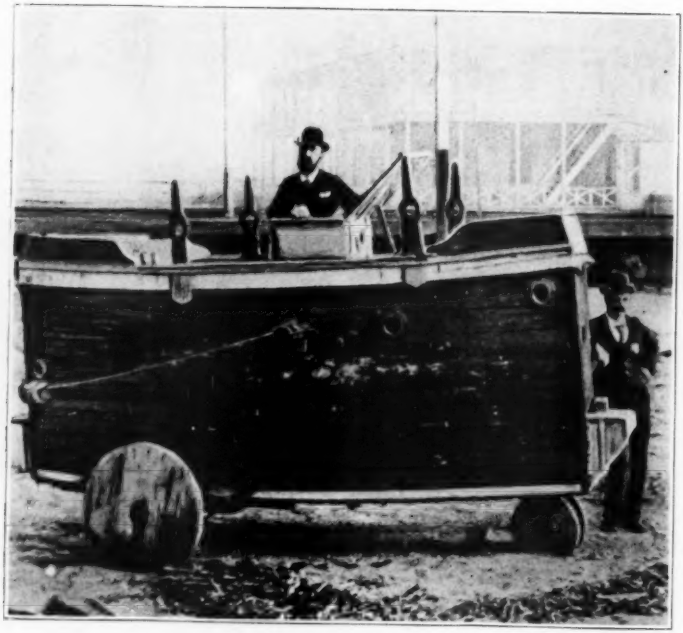
"ARGONAUT, JR."

The *Argonaut, Jr.*, was built in 1894–5, of yellow-pine timber painted with coal tar, and was 14 feet long, 4½ feet wide, and 5, feet high. She was propelled by hand power on a crank operating the driving wheels, her function being simply that of traveling along the bottom. Compressed air was carried in a soda-water tank, and a plumber's hand pump served as an air compressor.

In spite of this crudeness of construction, the *Argonaut, Jr.*, demonstrated the practicability of the diving compartment, her diving door having been opened at a depth of 16 feet. At this depth a diver was sent out, and the efficiency of the system was thus effectively demonstrated.

"ARGONAUT, 1ST."

This vessel, of steel, 36 feet long, and with 9 feet beam, was built in Baltimore, in 1897. She carried a 30-H.P. White & Middleton gas engine, a dynamo, air compressor, searchlight and



"ARGONAUT, JUNIOR"

"ARGONAUT THE FIRST."

"ARGONAUT THE SECOND."

water-ballast pumps. As in the *Argonaut, Jr.*, she was provided with a diving compartment, with geared wheels for movement along the bottom of the coast, and with a screw propeller for surface propulsion.

Although her size provided very little accommodation for the crew, as well as very limited carrying capacity, five men cruised over two thousand miles in her, on the surface and submerged, during 1898, in Chesapeake Bay and on the Atlantic Coast.

The *Argonaut 1st* is responsible for the development of the present large superstructure of the *Protector*, the extended cruising of the *Argonaut* having demonstrated the advisability of this construction for securing greater buoyancy in the cruising condition, with corresponding increase of seaworthiness, increased deck room and larger fuel capacity.

"ARGONAUT 2D."

During 1889 and 1900, the *Argonaut 1st* was reconstructed and became *Argonaut 2d.* As reconstructed, for wrecking purposes, she has a spindle hull 56 feet long, with 9 feet beam. She carries a buoyant superstructure wherein are installed air and gasoline tanks. She is 66 feet long, and, with a beam of 10 feet, her appearance in the cruising condition resembles that of a small schooner. She carries a 60-H.P. White & Middleton gas engine, dynamo, two air compressors, pumps, anchor hoists, telephones and a four-H.P. auxiliary engine. In her bow is a searchlight for illuminating the bottom when doing sub-surface work. She has accommodations for a crew of eight men, and an estimated cruising radius of 3,000 miles.

WHAT THE *PROTECTOR* OWES TO HER PREDECESSORS.

Her predecessors served to develop the following features, which have been incorporated in the *Protector:*

1. Diving door and compartment.
2. Superstructure.
3. Utilization of the bottom as a guiding medium by means of wheels in the bottom of the boat.
4. Control of depth of submergence, when not underway, by means of anchor weights.

5. Conning tower of considerable size.

6. An installation of gas engines which can be used while the boat is partly submerged.

7. Numerous less important details.

The important features of the *Protector*, which were not successfully developed and tested in her predecessors, are:

Control of depth of submergence by hydroplanes.

Storage-battery installation.

Omniscope.

Handling and firing of torpedoes.

"THE PROTECTOR."

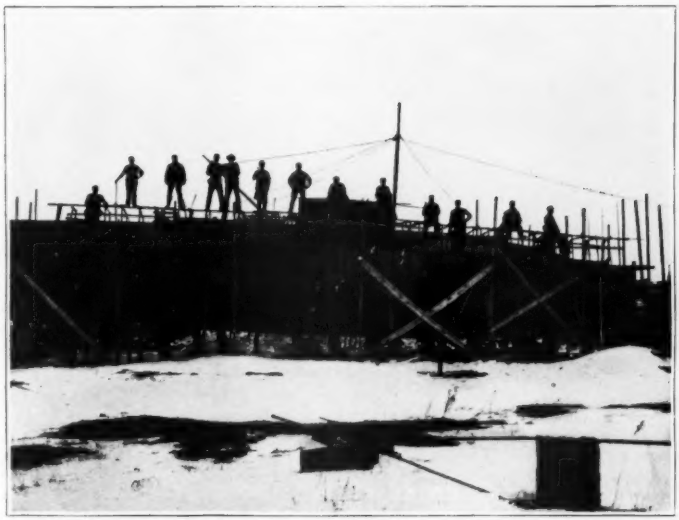
GENERAL DIMENSIONS.

Length over all, feet and inches	67- 6
Beam, extreme, over guards, feet and inches	14- 2
Depth from top of omniscope with wheels down, feet and inches	24- 8
Draught, light cruising condition, feet and inches	11- 9½
Displacement, light, tons	136.3
awash, tons	157.1
submerged, tons	174.35
Area midship section, cruising trim, square feet	102.32
submerged, square feet	147.2
Height of C. G. above base of keel, inches	83
C. V. above base of keel, inches	98
C. G. below C. V. (cruising trim), inches	15
deck awash, inches	9
submerged, inches	7.3
Height of C. G. above base of keel, superstructure full, ballast tanks empty, inches	93
Height of C. G. above base of keel, superstructure and conning tower full, ballast tanks empty, inches	96
C. V. aft of forward vertical, inches	371.4
C. G. aft of forward vertical (cruising trim), inches	374
C. V. (cruising trim), inches	3.16
Tons per inch immersion at cruising trim	1.3

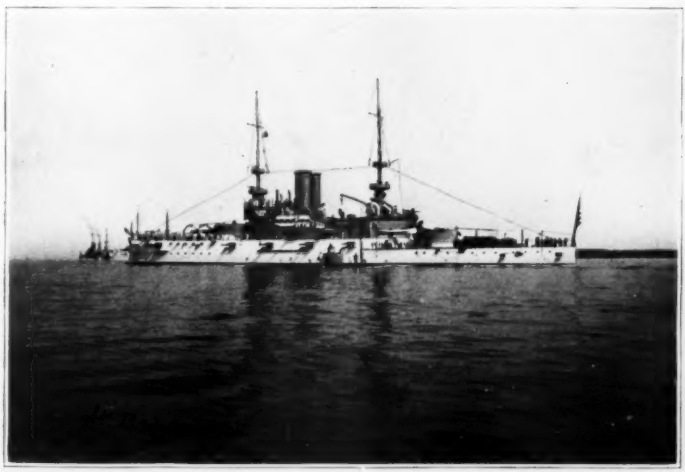
The following is as detailed a description of the *Protector* as can at this time be made public, and is necessarily very incomplete:

CONSTRUCTION.

The *Protector* consists of a heavy spindle hull of mild steel, 65 feet long, and with a maximum diameter of 11 feet 2 inches,



SPINDLE HULL, SHOWING FRAMING OF SUPERSTRUCTURE.



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A COMPARISON OF SIZES—"ALABAMA" AND "PROTECTOR."

with conning tower, designed to withstand an external pressure corresponding to a depth of 150 feet and carrying a light superstructure.

Hull.—There are thirty-six main frames spaced 20 inches apart. That portion of the spindle hull between frames No. 10 and No. 21 is cylindrical.

Hatch openings in the spindle hull at frames No. 8, No. 19 and No. 24, are respectively, 27¾ inches by 24 inches (elliptical), 22 inches by 24 inches (elliptical), and 32 inches by 32 inches (circular). Hatches are locked by sliding dogs operated by hand wheels and screws, providing great leverage.

Bow and Stern Castings. — Bow and stern castings are of malleable iron, conical in shape. The forward casting provides housing for the forward anchor, and the after one stuffing boxes for the vertical rudder post and operating shaft of horizontal rudder and a pocket for the pivot of the horizontal-rudder bell crank.

Keel.—The keel extends from frame No. 3 to frame No. 31, and has a depth of 17 inches amidship, tapering to nothing at the ends. It is worked in two side pieces or cheeks of steel, filled in, except at wheel pits and drop keel housing, with yellow pine, cut to fit, and bolted through.

Bulkheads.—At frame No. 6 is an air and watertight bulkhead designed to stand a test pressure of 75 pounds to the square inch.

Tank bulkheads are worked to the height of the floor plates. At frame No. 18, between the crew space and galley, is a frame carrying swinging doors, and at frame No. 20, between galley and engine room, the switch board serves as a partition bulkhead.

Superstructure.—A superstructure is provided for the following purposes:

I. To give buoyancy and seaworthiness in the cruising trim.

II. To provide stowage for fuel and air tanks outside of the spindle hull, for reasons of safety and economy of space.

III. To give deck room, thus contributing to habitability.

It is carried up from the greatest diameter of the spindle hull in a vertical direction, and is of comparatively light construction. Midway in the length is a watertight transverse bulkhead, which

divides the superstructure into two ballast tanks, fitted with suitable admission and discharge valves.

Access to within the superstructure is afforded through fourteen elliptical manholes, 16 inches by 12 inches.

Air and Oil Tanks—In the superstructure are carried:

Eight gasoline tanks of galvanized steel, with a combined capacity of 1,050 gallons, built to stand a hydraulic pressure of 100 pounds to the square inch.

Six high-pressure air tanks of 8-inch Mannesmann tubes, with a combined capacity of 21 cubic feet, built to stand a test pressure of 4,000 pounds to the square inch.

Two lubricating oil tanks, similar in construction to the gasoline tanks, with a combined capacity of 120 gallons.

Four low-pressure air tanks, similar in construction to the gasoline tanks, with a combined capacity of 12½ cubic feet.

All the gasoline tanks are connected to those on the same side of the ship, and all tanks on each side are connected with a copper service tank, fitted with gauge glass and float which automatically shuts off the supply. These service tanks are situated in the engine-room compartment abaft the engines.

All gasoline connections are of seamless brass tubing with brazed joints.

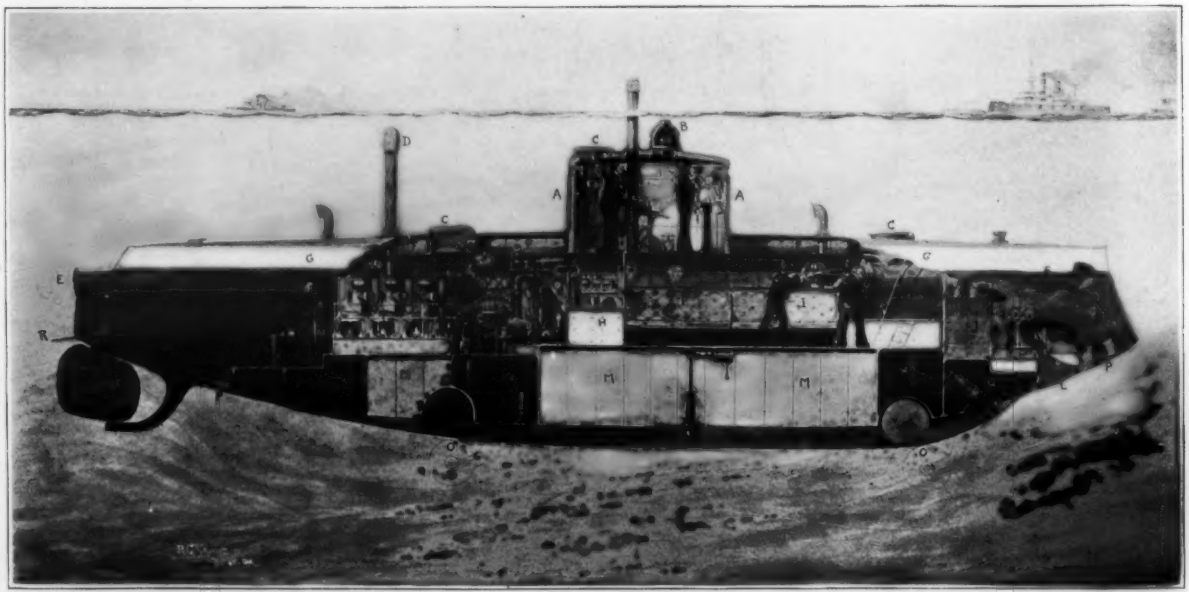
Connection between the high and low-pressure air system is made through a special Foster reducing valve, in the galley compartment, which reduces from 2,000 to 60 pounds.

Ventilators.—Ventilating trunks, consisting of cylindrical bronze castings, 6 inches in diameter, worked watertight through spindle, hull and superstructure deck, carry miter valves operated by hand wheels from within the spindle hull.

They are situated between frames No. 11 and No. 12, and between frames No. 27 and No. 28. Galvanized-iron cowls, 11 inches in diameter at mouth, screw on these in the usual manner.

Watertight bronze caps cover the valves when cowls are unshipped.

Deadlights.—The spindle hull is lighted through deadlights, in trunks, worked through the superstructure, and provided with clapper valves serving as battle ports.



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INBOARD PROFILE.

A, A, Bronze Conning Tower.
B, Sighting Hood.
C, C, Hatches.
D. Exhaust from Engines.
E, E, Torpedo tubes.
F, F, Gasoline Tanks.
G, G, Line of Spindle Hull.
H, Galley Compartment.
I, I, Crew Space
J, Air Lock.
K, Diving Compartment.
L, Diving Door.
M, M, Storage Batteries.
N, N, Drop Keel.
O, O, Wheels.
P, P, Anchor Weights.
Q, Q, Ballast Tanks.
R, Horizontal Rudder.

At a depth of 40 feet it has been found that the hull is reasonably well lighted, it being possible at this depth to read a newspaper under the deadlights without the aid of artificial light.

Torpedoes and Tubes.—The hull is pierced for three tubes with axes 3 feet 5 inches above center of spindle hull, two of which are forward, accessible from the crew space, with axes 4 feet apart, and one in the stern, opening into the engine room. In the light condition the tops of the tubes are awash.

The doors carry $\frac{1}{8}$-inch square rubber gaskets. The outer doors of the forward tubes house upward and inboard into a spectacle-shaped housing built into the superstructure.

The outer door of the after tube hinges at the bottom and drops to a horizontal position.

A firing pressure of 60 pounds from the low-pressure system is used.

The *Protector* is intended to carry 3.55 M. Mark III Whitehead torpedoes, three in the tubes and two spare ones (if considered desirable) under the berths in the crew space.

Torpedoes are taken aboard either by being floated into the tubes while the boat is in the light condition (which is the preferable method), or by removing the tails and striking them down the engine-room hatch. Within the boat they are handled by trolley straps hung from detachable rails in rear of tubes.

The boat's stability while submerged (both longitudinal and transverse) is such that the torpedoes may be loaded without materially affecting the trim of the boat or the control of depth of submergence.

Much trouble has been experienced in preventing corrosion of the shell of the torpedo furnished the *Protector*, due to the unavoidable dampness of the submerged tube. It seems very probable that in the event of a material increase in the number of submarines a torpedo with a bronze shell will be requisite.

Diving Compartment.—This forward compartment is, perhaps, the most spectacular feature of the boat. It is separated from the crew space by the air and watertight bulkhead at frame No. 6, and by an air lock, 42 inches long, built into this bulkhead,

and fitted with air and watertight doors—the whole being designed to stand a pressure of 75 pounds to the square inch.

The diving compartment is fitted with a connection to the low-pressure air system, a telephone, and a gauge with two hands, one of which registers the pressure due to depth of water, and the other the air pressure within the compartment.

In the bottom of the compartment, forming a portion of the hull, is a cast iron door, hinged at its after end and stiffened with transverse ribs, which, when the door is open, serve as steps.

It is raised by small hand winches, with wire-rope falls, and secured with dogs. Around the door is a coaming extended 30 inches above the hinge door.

To open the diving door when resting or running along the bottom, the air-lock doors being closed, a valve on the air connection is opened, and the pressure in the compartment allowed to increase gradually until the two hands on the gauge balance. A try cock in the door is then opened, and the balance of the pressures verified. The door is then undogged and allowed to swing open, giving, in clear water, a good view of the bottom.

A diving suit, fitted with telephone connection, is provided; the air hose being connected with a low-pressure air tank in the compartment, the pressure being controlled by the diver, through a valve in his helmet. Thus persons in the diving compartment are in telephonic communication with the diver and with the rest of the boat.

The diving compartment provides for:

I. Cable cutting.

II. Destruction of mines.

III. A means of escape for the crew, in case of total disablement of the boat.

IV. A means of communication with shore by cable when on picket duty.

In demonstration of these features the diving door has been opened five times in the writer's presence, in depths up to 32 feet. A member of the crew in diving suit and one in a bathing suit have been sent out, the latter to demonstrate the method of escape.

Wheels.—Two hollow cast-iron wheels, 34 inches in diameter, and with 10-inch rims, are hung by forged-steel jaws, pivoted through the cheeks of the keel at frame spaces No. 6 and No. 21½, and connected with hydraulic operating cylinders which are cushioned by pneumatic cylinders. The wheels house in pockets in the hull.

The pressure in the hydraulic cylinders, by which the wheels are raised and lowered, is obtained from a hand pump forward in the crew space. This installation is designed—

I. To permit the utilization of the bottom as a guiding medium. The conformation of the Atlantic Coast is, in many localities, such as to permit a submarine to run along the bottom well beyond blockading range without reaching her limit of depth (which, in the case of the *Protector*, is 150 feet). In a boat running thus, control of depth of submergence is unnecessary, and, being free from the influence of currents, a much straighter course can be steered than when running between the surface and bottom.

II. As complementary to the diving compartment in cable work, etc.

The cushioning of the hydraulic cylinders is intended—

I. To enable the boat to take the ground without injuring the hull. When submerged the *Protector* is always kept on nearly an even keel. In moderate depths (of less than 150 feet), with fairly smooth bottom, the commanding officer is thus relieved of a great deal of anxiety, inasmuch as a sudden descent to the bottom, due to loss of control, can have no harmful results. In submerged runs the *Protector* has frequently been allowed to take the bottom. This has always been accomplished without jar or shock.

II. To allow the boat to rest on the bottom in a ground swell, as in weathering a heavy storm.

Experience in the *Argonaut First*, while lying on the bottom off Cape Henry, demonstrated that at a depth of 30 feet, the ground swell was such as to lift and drop the boat 10 inches, with results disastrous to the machinery. The effect of this

drop, if not cushioned, is almost irresistible, inasmuch as the hull receives a thrust due to the weight of the water above it.

To utilize the wheels, the boat is given a slight negative buoyancy, causing her to rest on the bottom with a weight of only a few hundred pounds, thus allowing her to run over soft or irregular bottom. It is said that the *Argonaut* has climbed the 10-foot side of a channel dredged in clay. The wheels are in the nature of rollers, propulsion being as in the other stages of submergence. While running in this condition, if an observation is desired, the boat is brought to the surface by the hydroplanes.

Anchors.—The forward anchor is mushroom in shape, weighs 500 pounds, and is of grey iron, cast with an eye in its end, to which is secured a ⅝-inch steel-wire cable 250 feet long. The anchor hoists into a sheet-steel housing riveted to flanges on stem casting. The cable is led up through a wrought-iron hawse pipe over a pulley in superstructure deck, and through guides on deck to winch.

The after anchor is a dome-shaped grey-iron casting, weighing 500 pounds, and fitted with a cable similar to that of forward anchor. Its housing is a cast-iron dome fitted and riveted in the spindle hull between frames No. 32 and No. 33.

The anchors are hoisted by electric winches, operated from within the conning tower, the motors for which are in the after crew space.

The anchors, in addition to their use as such, serve as weights by which the control of depth of submergence is obtained while not underway. For this the boat is trimmed with a reserve buoyancy somewhat less than the combined weight of the anchors, and, with the anchors on the bottom, the boat is raised or lowered by paying out or heaving in on the cables. Thus, while at anchor in a submerged condition, as she might be under certain conditions of picket duty, the boat can rise for an observation and submerge at will.

Conning Tower.—A conning tower of considerable proportions is provided, for the following reasons:

I. To provide a buoyant moment well above the center of gravity, thus contributing to stability.

II. To localize the control when submerged.

III. To facilitate the surface navigation of the boat in comparatively rough seas.

IV. To provide a compass location well removed from the magnetic influence of the spindle hull.

The conning tower is 10 feet 1 inch long, 4 feet 7 inches wide, elliptical in plan, extending 6 feet 5 inches above the spindle hull, and is situated between frames No. 14½ and No. 20½. For a height of 33 inches the structure is of steel plate, the remainder being of Tobin bronze. All joints are machined.

The conning tower carries a sighting hood, omniscope, hatch and suitable lenses affording an all around view of the horizon.

Within is most of the controlling mechanism of the boat, namely:

Steering wheel for vertical rudder.

Operating wheel for hydroplanes.

Wheel controlling horizontal rudder.

Engine and motor telegraphs.

Telegraph controlling engines, motors and pumps.

Controlling gear of electric anchor hoists.

Valves for filling superstructure.

Air vents and air connections controlling filling and blowing out of superstructure and ballast tanks.

Gauges and dials as follows:

One 15-inch depth gauge registering to 75 feet.

One 6-inch depth gauge registering to 150 feet.

Dial showing inclination of hydroplanes.

Dial showing inclination of horizontal rudder.

Clinometer showing longitudinal trim.

Buoyancy indicator.

Water column showing level of water in superstructure.

When in cruising condition a detachable steering wheel and engine telegraphs are shipped on top of conning tower.

Compass.—In the dome of the conning tower, forward of the sighting hood, is worked a bronze compass case, carrying a Ritchie compass with a 7-inch double card, visible from above and below, and situated 3 feet 2 inches above the steel portion

of the conning tower, and 5 feet 11 inches above the spindle hull. This location, combined with the steadiness of the magnetic plane when submerged (the boat always remaining on nearly an even keel), contributes to sensitiveness and accuracy of the compass. The compass, as compensated, with but one quadrantal and no semi-circular correctors, has a maximum deviation of ¼ point.

Hydroplanes.—There are two hydroplanes on each side, flush with the superstructure deck, pivoted at frames No. 13½ and No. 22, and protected by a heavy steel guard, extending between frames No. 8 and No. 28.

They are operated in unison, through a system of cranks, shafts and bell cranks, from within the conning tower, and have a throw of about 15 degrees above and below the horizontal.

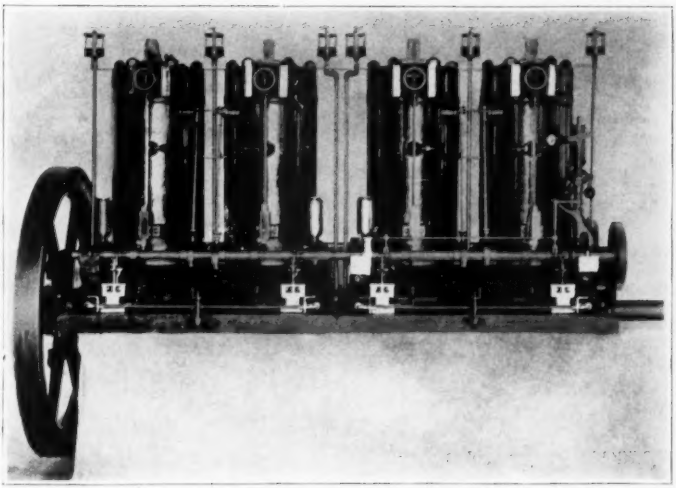
While running submerged, the depth of submergence is controlled by balancing the reserve buoyancy with the downward thrust, due to the depression of the hydroplanes.

Horizontal Rudder.—The horizontal rudder, of 10.8 square feet area, is carried on a steel shaft with babbited bearings in the stern casting and struts. It is controlled through bell cranks and operating shaft from the conning tower, and has a throw of 10 degrees in each direction.

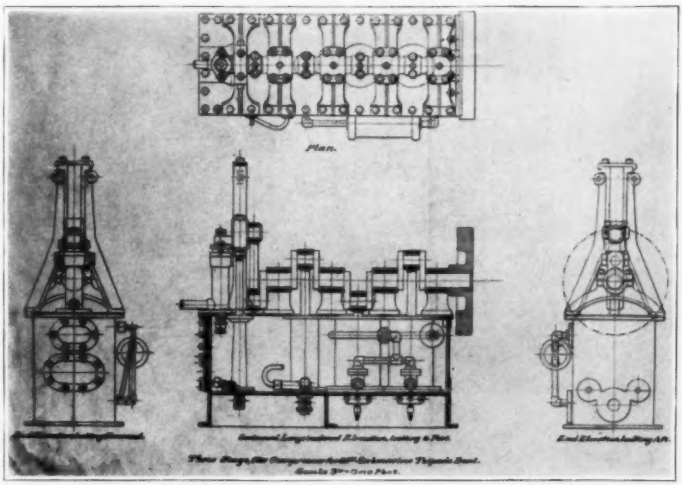
Drop Keel.—The drop keel is in two castings, of grey iron, with a combined weight of 10,000 pounds. Each casting is generally square in section, 17 inches by 17 inches by 7 feet 2 inches long. One end of each is curved to a 6-foot 6-inch radius, to give clearance in dropping, and the other ends are fitted to rest on sloping shoulders, worked in the regular keel for that purpose. The curved middle ends are hung by a T-headed bolt, worked in suitable pockets, and controlled by a shaft led up through a pipe and stuffing box to the level of the flooring in crew space. While submerged, a wrench is kept shipped on this shaft, a quarter turn of which releases the keel.

ENGINES.

There are two four-cylinder, four-cycle, White & Middleton gas engines, each of 120 H.P., on box-girder foundations, be-



FRONT VIEW OF GASOLINE ENGINES.



PLAN, END AND SECTIONAL ELEVATIONS OF COMPRESSOR.

tween frames No. 23 and No. 27, and with shafts 5 feet 6 inches apart. Cylinders are of 10-inch diameter, with a stroke of 12 inches, and, with the bases, are water-jacketed.

The ignition of the combustible, which is generally a source of trouble in engines of this type, is effected by:

I. A primary dry-cell battery.

II. A Holtzer-Cabot Co.'s magneto.

III. A current from the storage batteries.

With this combination, no trouble has been experienced due to lack of current.

Provision is made for using the engines in the third stage of submergence (with sighting hood out of water); the exhaust in this case being under water, and air for induction being taken through a water-excluding valve in top of sighting hood.

MOTORS.

Generator motors.—There is a 6-pole, shunt-wound Diehl motor in each main-shaft line, the armature shaft of which is connected to engine-crank shaft and to tail shaft by clutches. When only the engines are being used for propulsion, the armature revolves freely, with brushes lifted, and serves as a flywheel.

Each machine has a rated capacity of 37½ kilowatts at 125 volts, when driven at 300 revolutions per minute—a range of E. M. F. of from 80 to 160 volts, and a current capacity of 300 ampères at full load, with a momentary capacity of 450 ampères.

As in the case of the *Holland*, the ordinary armature insulation has proved insufficient, and the armatures have been rewound in the following manner:

The conductor, with its cotton covering, is baked to dry, dipped in varnish and baked twice, wound with three .01-inch thickness of red rope paper, wound with insulating tape, dipped in varnish and baked twice. After the armature is wound, the whole is again dipped and baked.

Originally, the conductor with its cotton covering, was simply insulated from the core by micanite laid in the slots, and, after winding, the whole was painted with Armalac.

The field coils, as a measure of precaution, were given a similar insulation.

Anchor-Hoist Motors.—In the after end of the crew space, on each side, in a 2-pole, series-wound Diehl motor, with a rated capacity of 2 H.P. at 110 volts, when driven at 1,450 r.p.m.

The armature shaft is vertical, and carries a worm at its upper end, through which it is geared to shaft of winch. Control of the motors is had from rheostats in conning tower.

BATTERY COMPARTMENT.

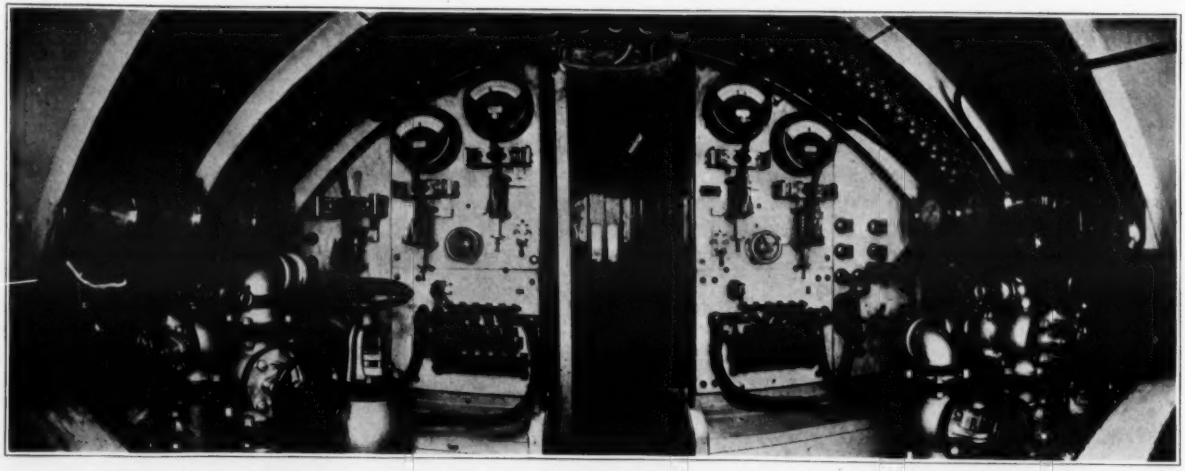
It has been well said that if one were to look for a location for a storage battery wherein obtained all the conditions to be avoided, it would be found in a submarine boat.

In installing batteries on shore, care is taken to provide a roomy, well ventilated, accessible place, free from moisture and from jar and shock. The difficulty in fulfilling these conditions in the limited space of a submarine boat, subject to rolling and pitching, is self evident, particularly when it is considered that on account of their excessive weight the batteries must be carried in the bottom of the hull, beneath the flooring.

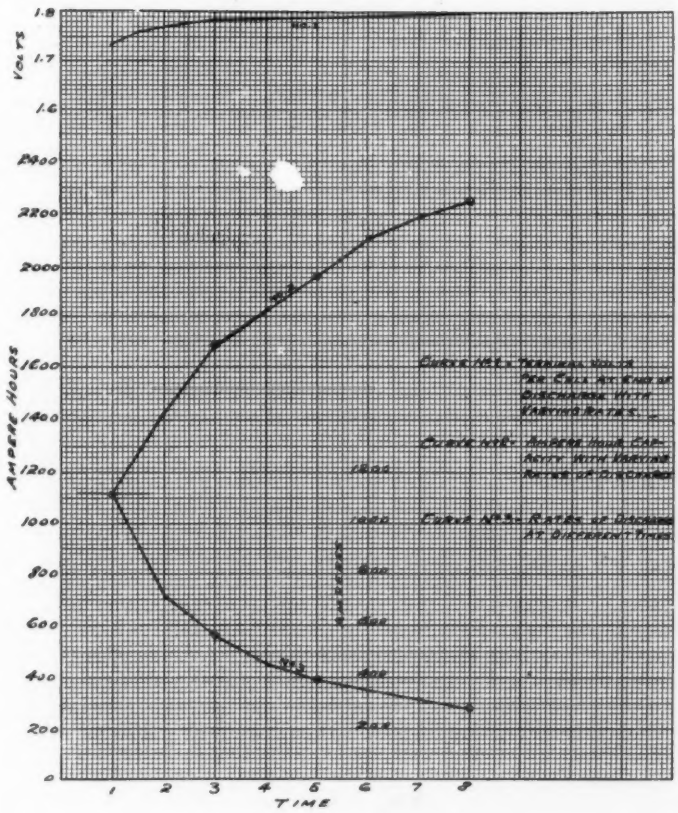
The storage-battery installation of any submarine, therefore, is of peculiar interest in that it represents a diffiult bending of conditions to suit a motive power, unsatisfactory as regards weight and space required, and endurance, but which, since it is the only power known which will give the required speed and radius without consumption of air, is a necessary evil in the submerged condition.

The storage-battery compartment of the *Protector* is mainly under the crew space, the floor of which is worked in transverse rabbeted strips of oak, alternate strips being easily removable for examination of cells. Side rows of cells are carried 12 inches higher under the transoms, the construction of base of transoms being similar to that of floor.

The bottom of the compartment is of creosoted pine, laid in concrete and covered with a special mixture of nonconducting cement. Wooden partitions are worked longitudinally to sepa-



SWITCH BOARD, PRESSURE GAUGES AND BALLAST MANIFOLD VALVES.



PERFORMANCE CURVES.
LAKE TORPEDO BOAT TYPE CELLS.

rate the rows of cells. Spaces between the cells are filled with asphaltum cement, for purposes of rigidity and insulation.

The compartment is suitably ventilated.

Storage Batteries.—There are sixty Gould cells, with spun plates of the Planté type, of the following general dimensions:

Weight of cell complete, pounds	1,198	
Total weight of battery, with asphaltum	75,895	
Overall dimensions of cell, inches	$35\frac{9}{16}$ by $18\frac{1}{4}$ by $16\frac{1}{16}$	
Number of plates in cell	17	
	Positive.	*Negative.*
Projected area of plate, square inches	837	837
Developed area of plate, square inches	8,931	10,422
Ratio of surface development	10.6	12.4
Estimated life of plate (in shore installation), years	5	$7\frac{1}{2}$ to 10

.....	8 hours.	5 hours.	3 hours.	1 hour.
Capacity of battery at varying discharge rates, in ampère hours	2,240	1,960	1,680	1,120
Electrical horsepower	42.5	59	85.5	...

The battery can be discharged at a rate well beyond the capacity of the motors without injury to plates.

The plates are electro-chemically formed, and are contained in cases of antimonious lead lined with lead. They are supported and insulated from the case by plate glass, and are separated from each other by hard rubber tubes. The top of each cell is provided with an antimonious cover, carrying a rubber gasket, in which is left a center opening 5 by $5\frac{3}{4}$ inches for terminal connections and to permit examination of cell and readings of acid density.

The installation is designed to permit an inclination of 45 degrees in any direction without spilling the electrolyte.

SHAFTS, CLUTCHES AND BEARINGS.

The main shafts are solid, of 3-inch machine steel, worked parallel, with bearings and clutches as follows:

At the after end of each engine, connecting the crank shaft with the dynamo shaft, is a 24-inch Frisbee clutch of special design. The hub of this clutch, $6\frac{1}{4}$ inches in diameter, is re-

ceived in the forward armature-shaft bearing, 4 inches long, at frame No. 28.

At frame No. 30 is the after armature-shaft bearing, 5½ inches long.

The armature shaft is connected to the tail shaft at frame No. 30½ by a Frisbee clutch similar to the other. On the forward hub of this clutch, on the port side, is the driving-gear wheel for the air compressor.

At frame No. 31¼ is a 6½-inch bearing for the tail shaft, with a cast-iron pedestal bolted to the floor.

All bearings are bronze bushed and babbited.

At frame No. 32 is a roller thrust-collar bearing of the following description:

A cast-bronze thrust block, 12 inches long by 8 inches wide, is bolted and babbited onto a built-up base, and carries two shaft supports spaced 5¾ inches apart. A cast-bronze cap in the form of a yoke bolts to this casting. The shaft supports carry bushings bored to a working fit with the shaft and threaded to permit the adjustment of a 6-inch roller bearing held by them against a split cast-iron collar 3 inches wide and 6 inches in diameter, clamped into a $\frac{1}{16}$-inch recess on the shaft.

From frame No. 32½ aft, the tail shaft is enclosed in a Tobin bronze sleeve ¼ inch thick, forming a part of the propeller pitch-adjusting mechanism. The forward end of this sleeve is closed by a stuffing box packed with hemp. The sleeve shaft pierces the hull through a cast-iron stuffing box riveted to the hull plating at frame No. 34.

PROPELLERS.

The propellers are four-bladed, 45½ inches in diameter, with iron blades cast with a 3-foot pitch. The developed area of each propeller is 6.8 square feet.

Provision is made for varying the pitch from 5 feet in the go-ahead motion to 5 feet astern, by hydraulic gear. This adjustment serves the following ends:

I. There is no gearing in connection of motors to driving shaft.

II. By decreasing the pitch the storage batteries may be charged while cruising at reduced speed.

III. By increasing the pitch the motors and engines may be used together for propulsion, giving 324 H.P. for higher speed.

IV. The speed may be controlled and the propellers reversed without the use of the motors.

These propellers have proved unsatisfactory, having developed a slip of 35 per cent. at full power.

This has been due assumedly to—

I. An insufficient diameter.

II. A relatively high projected area of blade, which, there being four blades, gives insufficient clearance, with corresponding hindrance to flow of water, between the blades.

III. Inaccuracies in pitch, due to warping of castings.

The longitudinal position of the propellers is such as to prevent a material betterment of the first condition.

New three-bladed propellers with blades of bronze, and of the following general dimensions, are being installed:

Diameter, inches	49
Cast pitch of blades, feet	4
Projected area, square feet	7⅞
Developed area, square feet	8¼

Air Compressor.—The compressor is located abaft the port engine, between frames No. 30 and No. 32½. It is designed to compress 60 cubic feet of free air per minute to 2,100 pounds, when running at 200 revolutions per minute. The working speed is 150 revolutions per minute.

The principal feature of its design is its compactness, it being 47 inches long, 22 inches wide and 34 inches high, except at high-pressure crosshead guides, which are 41 inches high.

It is driven from the port main shaft by gears which reduce the speed of the drive in the ratio of 2 to 1, and which consist of a bronze wheel carried on the hub of the after Frisbee clutch, and a cast-iron wheel on the face of the compressor flywheel, connected by a geared clutch of rawhide. The drive is taken from motor or engine.

The compression is in three stages, the first two of which are effected by trunk pistons; the high-pressure piston is actuated

by a connecting rod, whose crosshead is carried in guides bolted on top of high-pressure cylinder.

The cylinders are enclosed in a cast-iron tank forming a watertight chamber, with a continuation providing an airtight chamber below the low-pressure cylinders, and with flanged standards by which the compressor is secured to floor.

Air for induction is led from the highest point within the spindle hull to the airtight chamber in the casing. As a further precaution against water, three separators are provided.

A small circulating pump discharges the circulating water from the upper after end of casing overboard, the admission being at the bottom forward end, by gravity.

Handholes in the casing are located opposite the check valve of the circulating pump, and all induction and eduction valves.

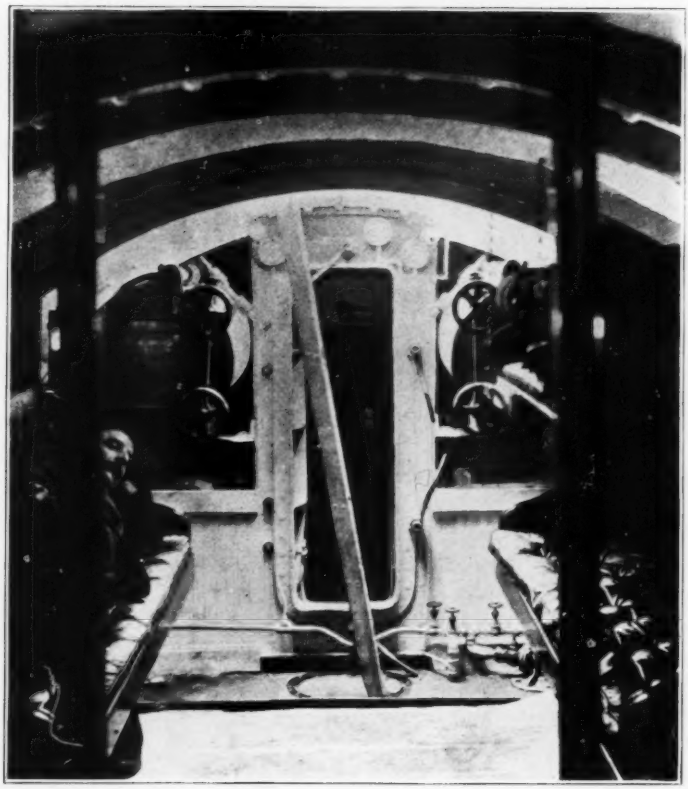
Ballast Tanks.—Ballast and storage-battery tanks are worked to the top of floors forward-and-aft, ranging from 6¼ inches below the axis of the spindle hull at frame No. 9 to 24 inches at frame No. 30. Manholes are provided for all tanks but one.

Indicators of the following description are to be installed in accessible and visible positions for all ballast tanks: A bronze box casting, 8 inches by 12 inches by 3 inches, with a glass face, contains a copper float at the end of a bell crank, the other end of which carries a rubber-faced valve seating on the air vent of the tank. When the tank is completely filled, water rises into the indicator tank and closes the air vent.

Manifold.—The manifold is in three castings of grey iron, situated above the floor plates, between frames No. 20 and No. 22. The sea connection is through a 7-inch globe valve on the port side, fitted with a strainer. There is a discharge on each side consisting of a 4-inch check valve steadied by a 7-pound spring.

The port side of the manifold has deliveries to the forward tanks and forward superstructure, and *vice versa.* All tank connections are of 4-inch pipes, except that of the after tank, which, for reasons of space, is 3-inch.

Pumps.—There are two special Rumsey rotary pumps, each with a capacity of 335 gallons per minute at 200 revolutions,



R. G. Skerrett, copyright, 1903.

Living Quarters—Looking Forward.

R. G. Skerrett, copyright, 1903.

Living Quarters—Looking Aft.

and a 4-inch suction and discharge. They are installed on the floor plates, forward and outboard of the engines on each side, and are connected to the main shafts by clutch gears which reduce the speed of drive in the ratio of 12½ to 11½. Thus at 300 revolutions (the designed speed of the engines) the capacity of each pump should be 462 gallons per minute.

The pumps can be driven either by motors or engines, and are noteworthy for their compactness, their extreme dimensions, exclusive of connections, being 29⅜ inches by 19½ inches by 14 inches.

Auxiliary Pumps.—Forward of the engines on each side, and worked by an eccentric of the crank shaft, is a single-acting vacuum pump, with a 6¼-inch plunger, packed with cup leathers, and a 1¼-inch stroke. This pump discharges overboard, and has a connection to the floor level of the engine room.

On the starboard side at frame No. 32½ is a small auxiliary air pump, worked from an eccentric on the main shaft, and designed to keep up the pressure in the low-pressure system.

In tanks No. 1, forward and aft, are brass pumps of 3-inch diameter and 4-inch stroke, operated by levers above the floor plates. They serve for adjustment of ballast and trim.

Forward in the crew space is a hydraulic-pressure hand pump with 1-inch suction, ⅝-inch discharge, 1¼-inch plunger, and 4-inch stroke, with connections to the hydraulic cylinders of both wheels. The suction connects with an oil tank installed for that purpose.

A handy billy, with a 1¼-inch suction and 1-inch discharge, is used for sweating out ballast tanks. A sea-delivery for this pump is provided so that it can be used in an extremity for emptying ballast tanks while submerged.

ACCOMMODATIONS.

The crew space, extending between frames No. 8 and No. 18, is finished in mahogany, and contains eight berths with pantasote cushions. The lower berths are in the nature of transoms, for which the upper ones, when lowered, form backs. Above and behind the berths are lockers. A folding table is provided,

the lower berths serving as seats. There are two electric heaters in this compartment.

Between frames No. 18 and No. 20 is the galley, with electric stoves and washroom. A water closet is located in the after engine room. An incandescent lighting system extends throughout the boat, there being 26 sockets.

SPEEDS.

There is no authentic data from which the speed of the *Protector*, with the old propellers, can be obtained. The following best speeds have been estimated from unofficial timings entered in her log, and are without tidal corrections:

Light condition, engines and motors, 9.47 knots.

Light condition, engines only, 8.57 knots.

Submerged speeds have never been taken.

With the new propellers the designer expects speeds as follows:

Light condition, engines and motors, 10 knots.

Light condition, engines only, 9 knots.

Partly submerged, engines and motors, 8 knots.

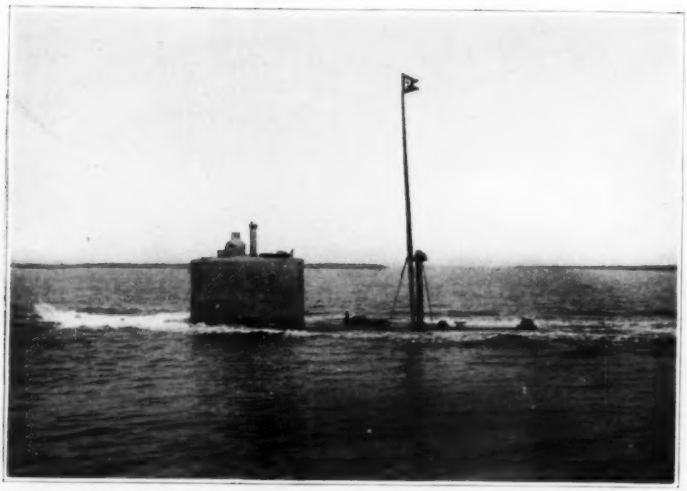
Completely submerged, motors, 6–7 knots.

CRUISING RADIUS.

No records of fuel consumption of engines have been kept The engines are guaranteed to deliver one horsepower for one hour on a pint of gasoline. From rough calculations made during the longer runs of the *Protector* the gasoline tank capacity should give a cruising radius of 350 knots in the light condition at full speed.

By filling two of the ballast tanks with gasoline, this may be increased to 1,000 knots. This may be still considerably increased by carrying gasoline in barrels in diving compartment or lashed on deck.

The submerged cruising radius with motors is likewise unknown. At full speed it is probably about 20 knots, and at an economical speed, about 30 knots.



R. G. Skerrett, copyright, 1903.

Degree of Submergence—Structure Awash.

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Degree of Submergence—Omniscope and Conning-Tower Hood Visible.

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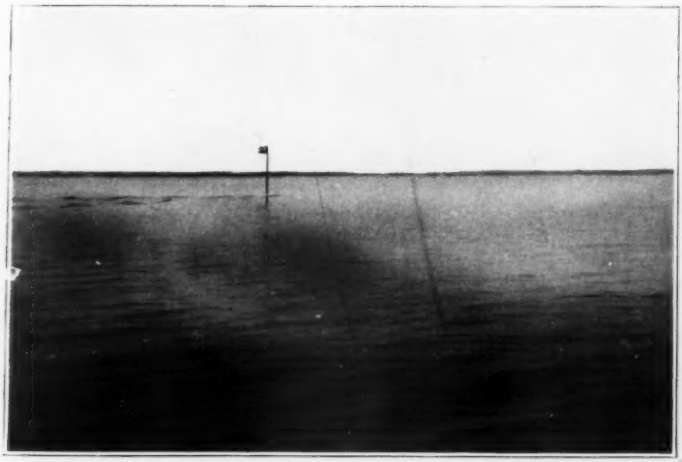
Degrée of Submergence—Omniscope Awash.



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Degree of Submergence—Signal Mast Alone Visible.

SUBMERGENCE.

The most remarkable feature in the performance of the *Protector* is the facility with which her depth of submergence is controlled, this being of particular interest, aside from its importance, in that it is one of the few qualities of the boat that were not developed in her predecessors.

There are five stages of submergence, viz:

I. The normal cruising condition, with superstructure and ballast tanks empty. In this condition the superstructure deck is about 16 inches out of water.

II. The war-time cruising condition, with superstructure filled and deck awash.

III. Superstructure filled and sufficient water in ballast tanks to submerge to the base of the sighting hood. This is the trim for submergence. A reserve buoyancy of about 280 pounds, corresponding to the volume of the sighting hood and omniscope, is maintained for all ordinary submergences. In this condition direct vision can be had through the sighting-hood lenses. While under way in this condition, by depressing the hydroplanes, two other conditions are assumed, namely:

IV. Submergence with nothing showing except the top of the omniscope.

V. Complete submergence.

To pass from the first of these conditions to the second requires about fifteen minutes; from the second to the third, about three minutes. Transition from one to another of the submerged conditions is almost at will, being well within the time required for an intelligent observation through the omniscope. This is, of course, of primary importance, inasmuch as when within sight of an enemy it is intended to run completely submerged, except for occasional verifications of bearing and range through the omniscope.

PROVISION FOR THE DISABLEMENT OF ELECTRICAL EQUIPMENT.

Propulsion for the first three stages is by engines and motors, singly or combined; in the last two stages, by motors alone.

The fact that in the third stage, by reason of an automatic in-

duction valve in the top of the sighting hood, admitting air for the gasoline engines and excluding spray and water, the engines may be used, gives the boat a large cruising radius in this condition, at comparatively high speed, and renders it likely that, under many conditions of sea, light and weather, the boat may get within torpedo range without being seen, in the event of the total disablement of her electrical equipment. In this case, of course, the omniscope would be housed, and the sighting hood, of a neutral color, could be discerned only with great difficulty. This feature assumes considerable importance when it is considered that the *elements most liable to disability in the submarine of today are the storage battery and electrical equipment.*

THE ADVANTAGES OF A DECREASE IN ELECTRICAL INSTALLATION.

It is intended in the near future to extend this advantage by taking the air for induction through the top of the omniscope, so that the engines may be used almost to the exclusion of the motors—certainly until within a mile of the enemy.

Could the electrical equipment of the submarine be removed in its entirety, the ability of the type to keep the sea without necessity of repairs would be increased three or fourfold, for it is only in this feature that the submarine is inherently short lived.

With the diminution of the necessity of propulsion by motors, as outlined above, a great step in the direction of endurance is taken. It is believed under this condition, that instead of storage batteries sufficient to give a cruising radius submerged of twenty or thirty miles, a radius of ten miles would suffice. Such a limited installation would serve for all the probable requirements of an engagement, and for the less important electrical requirements of the vessel, namely, propulsion on the bottom, as when engaged in cutting cables, and illumination.

An idea of the increase in speed and cruising radius to be effected by an increase in gasoline-engine installation at the expense of storage batteries, considering the question of weights only, may be obtained from the following rough calculation:

In the *Protector*, a weight of engines, fuel tanks and fuel, of 28,131 pounds, gives 8,400 H.P. hours.

A weight in storage batteries and motors of 89,895 pounds gives a maximum of only 340 H.P. hours.

That is, in the gasoline installation, a weight of 3.55 pounds suffices to develop one H.P. hour, whereas the electrical H.P. hour requires a weight of 264.4 pounds.

TRIM WHILE SUBMERGED.

When submerging with the hydroplanes, the horizontal rudder is set to keep the vessel on approximately an even keel, compensating any eccentricity of longitudinal trim due to distribution of weights. While running submerged there are no restrictions on the movement of the crew. Frequent tests have been made with varying numbers walking the length of the boat. Two men weighing 320 pounds, moving from engine room to air lock, while submerged, produce an inclination of 1½ degrees.

NO GREAT SKILL REQUIRED TO CONTROL DEPTH OF SUBMERGENCE.

The efficiency and safety of the *Protector* under submerged conditions may be evidenced by the fact that no great degree of skill is required in maintaining a uniform depth, since the writer on his first submergence was allowed to operate the hydroplanes for five minutes, and succeeded in maintaining a depth between 25 and 27 feet. The control of depth, while more delicate at the higher submerged speeds, may be maintained at almost any speed.

SAFETY.

In a circular issued by the Navy Department in April, 1893, prescribing requirements to be fulfilled by competitive designs for a submarine previously authorized by Congress, and which circular stands today as an authoritative and official expression as to the requirements essential in a submarine, the first quality demanded was that of safety.

The value of the submarine as a weapon of war will be vastly augmented if there can be implanted in the crews of such craft a spirit of confidence. The work of the crew serving in a submarine will ever be regarded as somewhat hazardous in char-

acter, and therefore all possible means should be taken to lighten the nervous strain which must accompany such duty. Service would be performed much more efficiently by those who are imbued with the belief that skill in the handling of the boat could be easily acquired, than by the crew who, in addition to the dangers of combat, see dangers inherent to the construction of their vessel, operating perhaps under conditions of sea to which they are unaccustomed.

MEANS OF COMING TO THE SURFACE.

The general safety of the vessel and crew is contributed to largely, as previously mentioned, by seaworthiness and stability. To provide certainty of return to the surface from the submerged condition there are the following series of expedients:

I. The reserve of buoyancy maintained while submerged.

II. Emptying of ballast tanks by air pressure.

III. Emptying of ballast tanks by power pumps.

IV. Emptying of ballast tanks by hand pumps.

V. Release of anchor weights, with combined weight of 1,000 pounds.

VI. Release of 10,000-pound drop keel.

VII. Escape of the crew through diving compartment.

VISION, ORIENTATION AND RANGE FINDING FROM THE SUBMERGED CONDITION.

The development of these features, which involve in so marked a degree the efficiency of the submarine while submerged, and particularly when in the critical position within torpedo range of an enemy, has in the past been so unsatisfactory as to affect seriously the ability to deliver a successful attack.

The problem of providing an instrument of this nature which will—

I. Project sufficiently above the hull to overlook the crests of seas without exposing any of the vessel;

II. Cover a horizontal field sufficient for the intelligent conning of the vessel;

III. Cover a vertical field sufficient to allow the observer to see the horizon during the changes of longitudinal trim to which the vessel is liable—

has been very difficult of solution in the case of the service submarines of the diving type.

THE ALTISCOPE.

In fulfilment of the first condition, the Naval Trial Board of the *Adder* and *Moccasin* recommended two lengths of observing tube, one to project 8 feet and the other 18 feet above the deck of the boat. Recent trials of these boats off Newport have demonstrated that, in a moderate seaway, the boats are controlled with great difficulty at depths less than 9 feet. Thus a long observing tube seems necessary to this type of submarine.

The internal diameter of the tube installed on these boats is about 3 inches. The dimensions of their conning towers prevent a material increase in this diameter. The result is that even with the shorter (8-foot) tube, the maximum field of vision without a reduction in size of image is about 1 degree 48 minutes. This means that without an increase of field by reduction in size of image (as in looking through the wrong end of a telescope), one could not see the whole of the broadside of a 500-foot battleship at a distance of 2½ miles.

A reduction in size of image has been adopted in these instruments to secure a reasonable extension of field. But, inasmuch as the human eye, accustomed as it is to seeing things in what we consider their true size, loses all sense of distance when a reduction or magnification of size is effected, it seems logical to expect that for a successful estimation of range, at least, the *commanding officer of a submarine must, in the absence of other means of range finding, be permitted to observe his target in its true size.*

The most natural method of increasing the horizontal field of an observing tube is, of course, by rotation of the tube in azimuth. This again requires a conning tower of sufficient size to permit the observer to follow the eye-piece in its rotation.

THE OMNISCOPE.

In the *Protector*, these difficulties have been anticipated by her stability and uniformity of longitudinal trim (she rises for an observation on practically an even keel), which—

I. Permit a comparatively short length of tube, inasmuch as the porpoising, peculiar to the diving type, is absent.

II. Provide against the loss of horizon due to great changes of longitudinal trim.

And the size of her conning tower, which—

I. Allows the use of a tube of large diameter.

II. Permits of rotation of observing tube to cover a large horizontal field, and to obtain orientation.

The omniscope consists of a 6-inch rolled-brass tube, led up from a convenient level in the conning tower to a height, when housed, of eight inches above the top of the sighting hood. It carries five upper prisms, with lenses. Four of these, used as finders, look ahead, astern and on each beam, each covering a field of 21¼ degrees, through a reduction in size of image. The fifth lens looks ahead and covers a field of 2 degrees 40 minutes, and presents the image in its true size, without distortion.

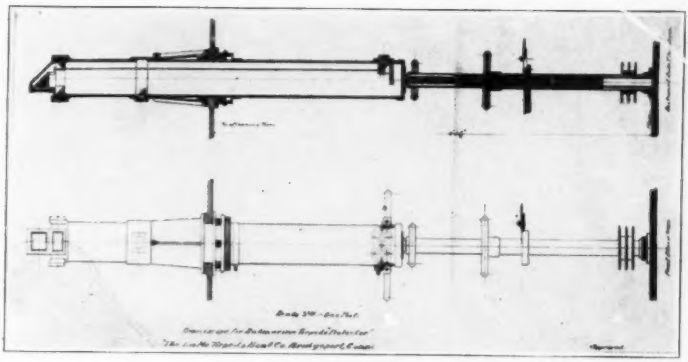
The omniscope is capable of rotation in azimuth to cover the entire horizon, and can be raised two feet. It works through a stuffing box packed with leather, and is given a 12-inch bearing to take the thrust while submerged.

Eye pieces and a lever for rocking the direct-vision prism to follow the horizon during slight changes of trim are provided at the base of tube within the conning tower.

On the front of the tube, level with the top of the sighting hood, are a lens and prism to indicate the submergence of the sighting hood, it being customary when rising for an observation to show only the omniscope.

An automatic installation keeps the helmsman informed of the bearing of the target, as seen through the omniscope.

On the lens of the direct-vision prism are two horizontal hair lines in the nature of a range finder. They are spaced to represent the height of a man at 300 yards, it being thought that in



ELEVATION OF OMNISCOPE.

R. G. SKERRETT, copyright, 1903.

CRUISING RIG OF CONNING TOWER.

making an attack a man will generally be visible on the deck of the enemy.

THE TRYING OUT OF THE *PROTECTOR*.

The *Protector* was launched at Bridgeport, Conn., on November 1st, 1902. Since December of that year she has been in active commission, under command of her designer, with a crew of seven skilled men and a steward. This time has been given up to the solution of the many unforeseen problems of detail incidental to the presentation of a new type of submarine construction, and to verifications and unofficial demonstrations of the capabilities of the boat.

Since July 9th, 1903, the writer has been on board during all runs and submergences. Of these, most have been short runs out into Long Island Sound for submergence and torpedo trials. Submerged runs have been made at depths up to 48 feet, in depths of water to 16 fathoms, and in seas varying from smooth to moderately rough. In a rough sea there is some difficulty in trimming the boat for submerging, due to fluctuation of water level in bouyancy gauge, but once submerged there is no difficulty in handling her under the conditions thus far experienced.

The following surface runs have been made under her own power:

From Bridgeport to New London, ...	65 knots.
New London to Newport, ...	48 knots.
Newport to Bridgeport, ...	98 knots.
Bridgeport to Port Jefferson and return, .	28 knots.
Oyster Bay and return, . .	50 knots.
Elizabethport, N. J., and return,	120 knots.

The only tender used by the *Protector* has been a small gasoline launch towed astern, and used as a running boat.

ENDURANCE OF THE GASOLINE ENGINE.

All of the runs were made without a stop or delay of any kind due to trouble with the motive machinery, except that during one trip one of the eight cylinders of the gasoline engines failed to function on account of water in the induction pipe.

The marked simplicity of gasoline engines for such propulsive

work, since such installation obviates the use of steam boilers and condensers, leads to the possible conclusion that the day may not be far distant when we may possess submarine cruisers, with an endurance of machinery afloat that is not possessed by the present high-speed surface torpedo vessels.

HABITABILITY.

While cruising the crew have always eaten and slept on board, the meals being cooked without difficulty on electric stoves. In all respects, save that of deck room, und under all conditions of weather, this boat provides greater comfort for the crew than the surface torpedo boats on which the writer has served.

DEDUCTIONS FROM THE PERFORMANCES OF THE *PROTECTOR*.

Those most familiar with the construction and operation of the *Protector* regard her as an advanced design in the development of the successful submarine. The results secured show that this type of construction possesses qualities capable of elaboration, and that further increase in efficiency can be expected. The possibilities of the boat have been understated rather than overestimated.

It is safe to say the *Protector* has demonstrated—

I. The possibility of submerging with facility and at will without dangerous destruction of longitudinal stability.

II. The ability in a submarine to combine seaworthiness and habitability to such a degree as to permit her to keep the sea for long periods unattended.

A study of the boat suggests—

I. That the logical development of the type will be a submersible cruiser of greater size and cruising radius, with a surface speed sufficient to enable her to run down her prey, and to cruise with a battle squadron.

II. That, if the development of the storage battery is not such as to greatly decrease the weight per unit of horsepower and its liability to derangement, the electrical installations of the future in submarine vessels will be proportionally much less extensive than at present.

REPORT CONCERNING THE DESIGN, INSTALLATION AND OPERATION OF THE TURBINE ENGINES OF S. S. *REVOLUTION.*

DEPARTMENT OF THE NAVY, BUREAU STEAM ENGINEERING,
WASHINGTON, D. C., *October 6, 1903.*

Sir: The undersigned, having been appointed a Board to witness the performance of the steam turbines installed in the Steamer *Revolution*, submit the following report:

1. The turbines above referred to are of the Curtis two-stage, compound, marine type, reversible and condensing. They are two in number, directly connected to the thrust and propeller shafting, each set of turbines driving one screw of about 4 feet 6 inches in diameter and 3 feet 4 inches pitch. This plant, being the first large set of steam turbines installed for screw propulsion in the United States, it seems appropriate to mention that the general performance of these turbines, although themselves built rather in the nature of experimental appliances, was quite satisfactory, and they are well adapted as a type of motor for the transmission of propulsive power.

2. Aside from the ordinary advantages claimed for the steam turbine, the most noticeable feature observed was the almost entire absence of vibration or disturbing noises usually attending the running of fast-moving reciprocating engines, and the slight care and attention needed while in operation. These turbine engines were easily started, stopped and reversed, the time required for each operation being only that necessary for the opening and closing of a set of valves connecting the turbine casings with the main steam pipes. Five seconds was the maximum time consumed for the operation of reversing the motors from full speed ahead to full speed back, with two men handling the gear, one to each turbine. This may, however, be done by one man

at a slightly increased expenditure of time. Various runs, with corresponding stopping, backing and turning of the ship, were made at different speeds of rotation, the maximum speed of engines being about 650 R. P. M., and the minimum about 250. At full speed ahead the vessel turned readily either to starboard or to port in about thrice her length. Dead in the water, and starting one propeller full speed ahead and the other full speed astern, the vessel turned in her own length. Dead in the water, and with the helm hard to starboard, and then going ahead on the port propeller, the vessel turned readily to port, due to the action of the propeller race upon the rudder. Under these latter conditions it was noticed, however, that the vessel ceased to turn when the wind was well on the port beam.

3. Running full speed ahead with both screws, and then suddenly reversing the engines, the ship was brought to a standstill in about 32 seconds; the same conditions with one screw, required about 38 seconds. Similar operations were repeated with the vessel running at lower speeds, and about the same amount of time for reversing was required. While there was no perceptible vibration to the hull amidship or forward, there was an appreciable amount directly over the screw propellers, probably due to want of a more perfect balance in the propellers.

4. The several trials not being carried on with a view to ascertain any data of efficiency or economic results of steam consumption, the only information obtainable that would convey any idea regarding the economic efficiency of the plant is an abstract from Prof. Jas. E. Denton's report on the Curtis Steam Turbines for the Yacht *Revolution*, in which report it is stated that at 672 R.P.M. an equivalent power of about 1,800 I.H.P. was developed by both turbines at a steam consumption equal to 18.14 pounds per equivalent I.H.P. This is, however, believed to be excessive, owing to the use of the steam in two stages only. With four-stage turbines, however, the consumption of steam per I.H.P. should be reduced well within the margin of prevailing averages for marine engines. A quite noticeable occurrence was the sudden drop in the steam pressure at full powers, indicating plainly the inability of the boiler installation to furnish the requi-

site amount of steam needed for the efficient working of the turbines.

5. The general installation of the machinery afforded sufficient room for the proper care and handling of the motors. Some of the auxiliaries, however, were, by necessity, placed in somewhat inaccessible places. Thus the air pumps, circulating pumps and main condensers were located below the working platforms, which is a natural outcome of the considerably higher position of said platforms relative to the hull, due to the fact that the center of the turbine shafts must have a much higher position than with ordinary engines. The forced-draft blowers, six in number, and all on one shaft, were driven by a small Curtis turbine, running at about 2,800 R.P.M. The generator in the electric plant was similarly driven.

6. The clearance spaces between the stationary and running blades on the revolving drum being made extremely small (from .02 inch to .06 inch), the amount of which is ascertained by a small, simple, micrometer attachment, and there being no flexible coupling arranged for between the turbine shaft and the thrust shaft, the bearing of the latter was provided with a handwheel mechanism actuating a yoke, so arranged as to maintain the original clearances by sliding the thrust-block on its sole plate, propeller, shafting and turbine drum all moving together. The wear on the thrust-block being very slight, this adjustment is only rarely resorted to.

7. The turbine shaft bearings, one at each end of the casing, run in oil, which is continuously circulated and cooled by water coils connected with the oil pump, no water circulation being used direct on the journals; water circulation is, however, used in the thrust bearing.

8. The exhaust from each turbine issues from the casing of the upper half of the secondary stage through a 20-inch pipe into the condenser. The condensers are the ordinary type, with about 1,100 square feet cooling surface each. They are fitted with bottom-scoop injection, assisted by a small circulating pump. The air pump is a double Blake featherweight, 6 inches by 12 inches by 8 inches, placed forward of the turbines and

partly below the working platform. A vacuum of 28 inches is maintained without trouble. Other auxiliaries consist of an auxiliary condenser and feed-water heater, one Blake feed pump, 9 inches by 6 inches by 8 inches; one Blake compound feed pump, 6 inches by 9 inches by $3\frac{3}{4}$ inches by 8 inches; one Blake duplex oil pump, 2 inches by $1\frac{1}{4}$ inches by $2\frac{3}{4}$ inches; one Knowles sanitary pump, $2\frac{1}{2}$ inches by $1\frac{1}{2}$ inches by $2\frac{3}{4}$ inches; one Blake auxiliary air and circulating pump, 4 inches by 5 inches by 5 inches by 5 inches; and two Seabury circulators, 4 inches by $4\frac{1}{2}$ inches. All of these auxiliaries are placed in the engine room with the exception of the auxiliary condenser, which is under the boiler-room floor.

9. The boiler plant consists of two Seabury double-ended water-tube boilers of 94 square feet of grate surface, and are arranged to be operated under forced draft.

10. The main turbines consist principally of the casings, nozzles and revolving drums containing the bucket wheels and shafts. The steam enters at a pressure of 250 pounds through four steam nozzles, each of which supplies one-fourth of the full power of the turbine, and acts consecutively upon a series of bucket wheels constituting the "first stage," in which the steam expands down from 265 pounds absolute to about 16 pounds absolute. It then flows through four other nozzles, to act upon another set of bucket wheels, which form the "second stage," and there expands from 16 pounds absolute to less than 1 pound absolute, with condenser vacuum of 28 inches, the work done by the steam upon the buckets being thereby nearly equalized.

11. Reversing is made possible in a very simple manner by placing vanes on the outer rim pointing oppositely to the ahead buckets, all enclosed in the casing of the secondary stage. A separate steam pipe leads to this part of the casing and the steam is admitted by a separate valve.

12. The general ease with which the machines in question are cared for and handled is undoubtedly due to the great simplicity of the mechanism when compared with the reciprocating engine. As an engine it requires but little previous "warming up;" no cylinder drains to be opened nor any general filling up of oil

boxes, sight feeds and wick cups, the turning on or the shutting off of the main steam supply being the principal and almost the only operation remaining. It might be here incidentally mentioned that the above points are not features peculiar only to the turbines of which this report treats, but hold in common for steam turbine machinery of the various designs and makes. The space occupied, judged in the light of a casual observation, is no less, but rather more, than that required by engines of the torpedo-boat class; but, as there is not the same amount of overhauling necessary, the extra space can well be spared. The weights of the turbines proper in this case are alleged to be 8¾ pounds per equivalent I.H.P., and are thus less than those of torpedo-boat engines, which run about 11½ pounds per I.H.P. The attention required and the amount of oil used is extremely small, owing to the entire absence of any internal or external oiling except the main bearings, which are automatically oiled. No frequent adjustments are required, as there are no rubbing surfaces, which may best be illustrated by the fact that the turbines in question have never been apart since first put up, covering a period equal to one and one-half years.

13. The points in conjecture with regard to these machines being their efficiency and their capacity to compete in economy with the best marine engines, should be ascertained by more accurate tests, both on land and on board ship, where this is possible. The opinion of the undersigned is that such tests are warranted, and eventually will result in the application of a motor for screw propulsion highly satisfactory to the Naval Service.

14. Additional fore-and-aft space will certainly be required for any efficient installation of turbines as compared with the reciprocating engine installation, and particularly will this be found in torpedo boats. This fact should be well recognized and be given due consideration, so that all parts of the mechanism, but particularly the condenser and other auxiliaries, may be installed in such manner as will permit frequent and necessary overhauling from the engine room.

15. As the turbine is more dependent upon a high vacuum than is the case with a reciprocating engine, it is essential that

the main condensers and main air pumps must be favorably placed in the ship for ready examination and overhauling.

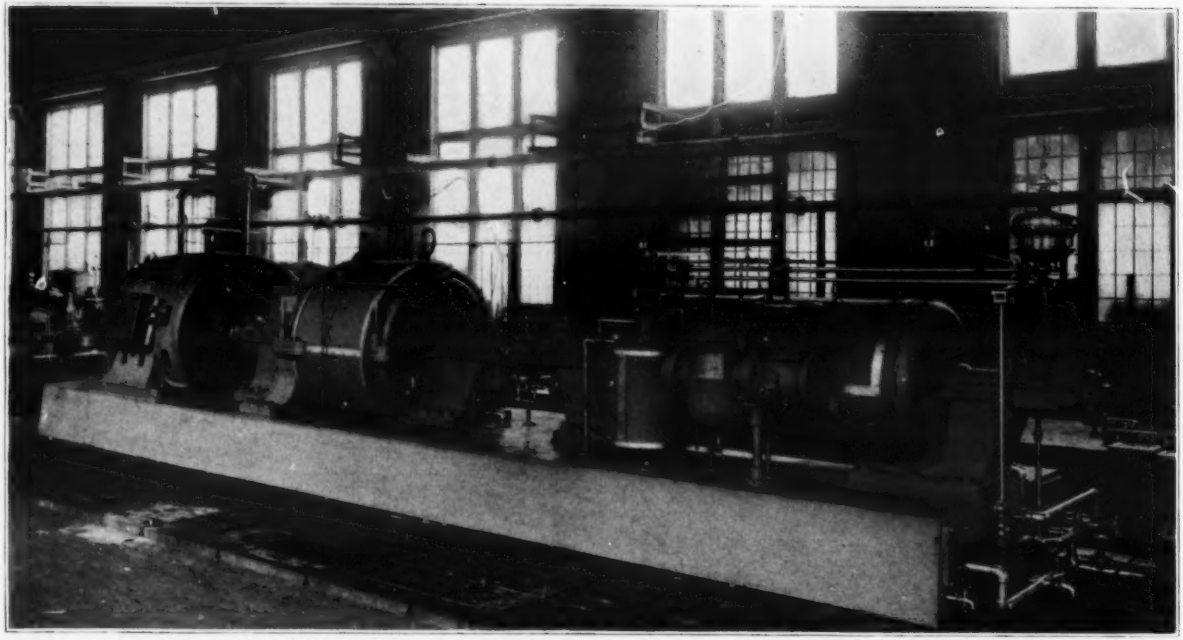
16. In order to maintain the requisite vacuum it is also essential to efficiency that the auxiliary appliances should not exhaust into the main condenser. It will thus be found that the auxiliary condenser, with the adjunct auxiliaries, will have to be larger than in the case of the reciprocating engines.

17. It is an absolute necessity that both the main throttle and main reversing valves be perfectly tight, and thus it may be imperative to design a special form of valve for such purposes, so that frequent overhauling and grinding in may not be necessary.

Very respectfully,

A. B. CANAGA, *Commander, U. S. N.*
J. R. EDWARDS, *Lieutenant Commander, U. S. N.*
W. M. PARKS, *Lieutenant Commander, U. S. N.*

The CHIEF OF THE BUREAU OF STEAM ENGINEERING.



WESTINGHOUSE—PARSONS STEAM TURBINE, FOR OPERATING 1,000-K.W. GENERATORS.

REPORT OF BOARD TO OBSERVE AND REPORT CONCERNING THE EFFICIENCY OF TURBINE ENGINES.

WASHINGTON, D. C., *July 27, 1903.*

SIR: The undersigned, having been appointed to witness the trial and to take data of certain turbine engines designed and built by the Westinghouse Machine Company, submit the following report:

2. The trial above referred to was made with one 1,000-k.w. steam turbine built for the Cleveland, Elyria and Western R. R. It was of the two-stage type, direct connected to the revolving field of an alternating-current generator, 400 volts, 2-pole, 3-phase, 3,000 alternations, the general appearance of which is shown in the half tone. It having become a commercial necessity in this country to build alternating-current generators to run averagely at 25 per cent. overload, and to make them to stand 50 per cent. for an hour or two, the turbine in question, therefore, if it were used for driving anything but an alternating-current generator, would be rated as a 1,500-k.w. turbine instead of 1,000-k.w. turbine.

3. The general design of the turbine in question is, with the exception of various improvements in details, substantially the same as that which Parsons has for a number of years used in stationary-engine practice, and with some modifications has also installed for marine use. Its manufacture has been carried on by the Westinghouse Machine Company for some years past, and as a sample of the applications, to which this machine has been successfully made, the four 400-k.w., 3,600 R.P.M. turbines installed at the Westinghouse Air Brake Company may be mentioned; these machines have been running continuously for about four years.

4. The original design, having called for a reheater between the H.P. and the L.P. turbine, was the cause of the great length of

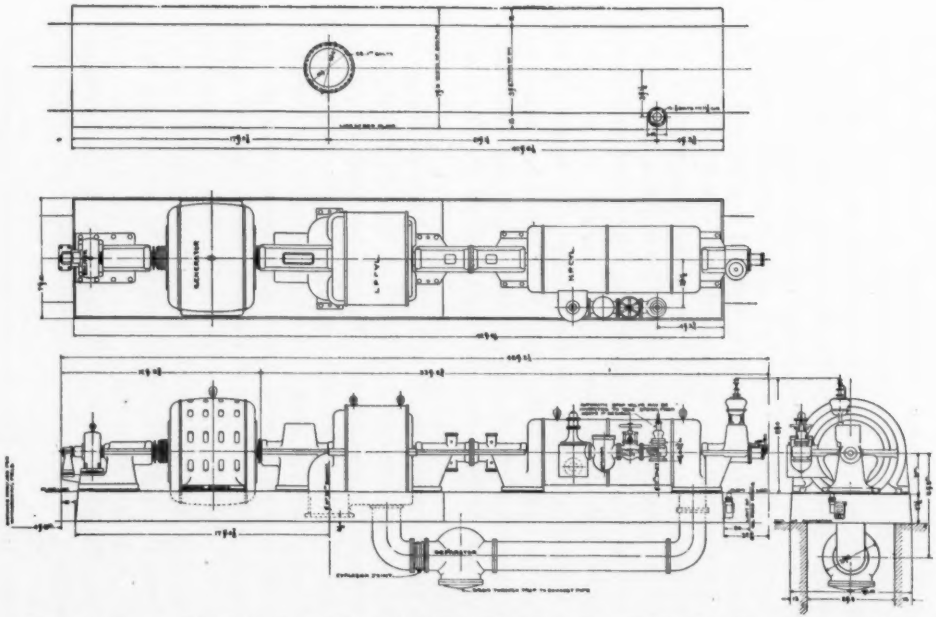
this turbine (about 43 feet 3 inches, including the generator and its bearings), and without said reheater the cylinders may be brought close together and thereby considerable space saved. The reheater was not used in this machine (and is not likely to be in others), but substituted with an ordinary receiver pipe and separator. The two turbine cylinders and the generator are all bolted down to the same bedplate, which is made of considerable depth and is quite heavy, approaching 33 per cent. of the weight of the whole outfit. This bedplate is not bolted to the foundations.

5. The testing of this machine was performed in the Turbine Department and was carried out with great accuracy, being a part of the plant specially built for the purpose of testing all turbines. Steam was supplied from the boilers of the general plant of the Westinghouse Machine Company, and entered the casing of the H.P. turbine through a series of valves. There is first an automatic stop valve, then a throttle valve which connects with a steam strainer placed between this throttle valve and the admission valve shown in plate I. This valve is composed of a double poppet, actuated by a piston and a small relay valve, which receives its reciprocating motion from levers connected by an eccentric through gears to the turbine shaft.

6. The movement of the relay valve is controlled by a ball governor, and the steam is by this manner of regulation admitted in puffs instead of continuously.

7. The exhaust was taken out at the bottom of the L.P. turbine and led through a 30-inch exhaust pipe to the bottom of an ordinary surface condenser. The lower part of this pipe was formed into a hotwell, at the bottom of which was placed a nozzle with a by-pass valve, connecting the suction pipe on the pump either direct to the condenser or to the hotwell. A gauge glass, with means provided to ascertain the water-level line, gave readings of the amount of water contained in the hotwell when beginning and finishing the trial. The hotwell pump discharged directly into either one of the two weighing tanks placed on scales to ascertain the amount of water condensed.

8. The circulating pump was driven by a direct-connected



DESIGNED AND BUILT BY THE WESTINGHOUSE MACHINE CO., PITTSBURG, PA., U. S. A., FOR THE CLEVELAND, ELYRIA & WESTERN RAILWAY.

1,000 kw. steam turbine. 150 pounds boiler pressure. 1,500 revolutions per minute. Condensing. Direct connected to 1,000 kw. revolving field A. C. generator, 2-pole, 3-phase, 3,000 alternations, 400 volts.

PLATE 1.

Westinghouse Standard engine, running at about 360 turns per minute. The air-pump suction was attached at the top of the condenser and connected to an Alberger dry-vacuum air pump. As the name infers, no water is drawn out by this pump, but merely air and vapor. The air pump itself is composed of two tandem cylinders, and is worked at a uniform speed of about 80 revolutions per minute, maintaining an average vacuum of about 27½ inches. The near ends of the cylinders are connected by means of a pipe, the other end of each cylinder being joined to the condenser and the atmospheric exhaust respectively. A suspicion existing that the vapor discharged contained some amount of water, this atmospheric connection was coupled up with a small condenser, with the result that about 44 pounds of water was separated per hour. This should be added to the results put down in the test tables.

9. The water separated in the receiver pipe was measured at every interval of readings and trapped back to the condenser.

10. After passing through transformers, the current was absorbed in a water rheostat, and readings obtained on the usual instruments.

11. The turbine was started at 9 A. M., gradually working up to full load. More than one hour being required to sufficiently heat the water in the rheostat to obtain the necessary resistance to the current, the actual trial was not begun before 10·15 A. M., of which the various test data for the first hour are included in folio A. The number of revolutions of the turbine was given by counting the strokes of one of the governor levers, and then by multiplying same with 9⅓, which is the ratio between the worm on the turbine shaft and the gear actuating the governor levers. From the table, the revolutions will be seen to vary between 1,480 and 1,456 per minute.

12. During the second hour of the trial the turbines developed approximately 1,500 k.w., all references of this test being contained in folio B.

13. During the third hour the load was working down from 1,500 k.w. to approximately 1,000 k.w., the data of which are contained in folio C.

14. During the fourth hour the turbine was running approximately at its rated capacity of 1,000 k.w., the test data of this trial being found in folio D.

15. The general results contained in the folios referred to above will be found in the table at the end of this report.

16. It is to be noted that during the whole run of the four hours the revolutions of the turbine shaft were nearly constant or 1,500 turns, although the load was increased from its rated capacity of 1,000 k.w., to 1,500 k.w., an overload of 50 per cent. By comparing the pressure at the H.P. inlet, when working under the two different loads, due entirely to the action of the governor, the above-mentioned fact may readily be accounted for.

17. The turbine ran without noise, the humming discernible being due to the generator. Its revolving parts, being accurately balanced, ran perfectly smooth, without even the slightest vibrations. All end thrust due to an excess of pressure on the steam side of the moving blades was counteracted, as is usual, by balance pistons, the difference in thrust, when finally adjusted, being so small as to allow the drum to be moved in its axial direction by a slight touch of the finger. Compared with a reciprocating engine there are fewer parts that need watching, and a smaller engine-room force will in all likelihood therefore suffice.

18. The space occupied by a battleship engine of the usual stroke and piston speed is approximately 0.75 cubic feet per I.H.P. The space required for the turbine (bringing the cylinders close together and eliminating the generator) is about 0.68 cubic feet per I.H.P., the above being figured on a basis of efficiency of 0.85 in the reciprocating engine.

19. The weights of the turbine may be cut down to some extent by substituting steel castings where now cast iron is used, for instance, in bedplate and casing. The revolving drum and shaft is made of forged steel. The grooves containing the moving blades are slightly dovetailed, each blade and distance piece, after being dropped in place, is subjected to pressure sufficient to upset the end and thus fill out the dovetail, thereby firmly gripping the blades. For the stationary vanes the grooves are parallel. The edges are caulked after the blades are all assembled.

20. The free ends of the longer blades are lashed with brass wire, then soldered, forming a sort of shroud or continuous band at the outer end. The blades are made from brass material, drawn in long lengths to actual section required. They are afterwards sawed off to any length desired.

21. The bearings, only having to sustain the weight of the revolving drum with its accessories, are barely loaded to a pressure of more than 50 pounds per square inch against from 300 to 400 pounds in a reciprocating engine; the peripheral velocity is, however, great, amounting to about 50 feet per second, against about 10 feet for battleship engines.

22. In conclusion, it may be said, that for the purpose the steam turbines now are being used, viz: for driving electrical generators, they are admirably suited, and are undoubtedly, both as a mechanism and a heat engine, in some cases at least, superior in efficiency when compared with the ordinary steam engine.

23. When analyzing the figures given below, it should be borne in mind that the turbine cylinders were not lagged, but simply covered over temporarily with felt roughly tied in place, and, therefore, the results obtained may, under other circumstances, be slightly improved upon. The table appended below will present a comparison of the horsepower in designating the capacity of motors when used for driving electrical machinery and for ordinary or marine purposes, and will be for the turbine in question:

Kilowatts, ...	1,510	1,019
Equivalent electrical H.P., . . .	2,023	1,365
Equivalent brake H.P. delivered to the generator, at 95 per cent. efficiency of the latter, ...	2,131	1,437
Equivalent indicated H.P. of a reciprocating engine at 85 per cent. combined efficiency of gererator and engine, ...	2,380	1,606
Revolutions per minute, ...	1,456	1,480
Steam pressure per gauge above throttle, .	149.4	146.2
Steam pressure per gauge below throttle, .	148.8	101.1

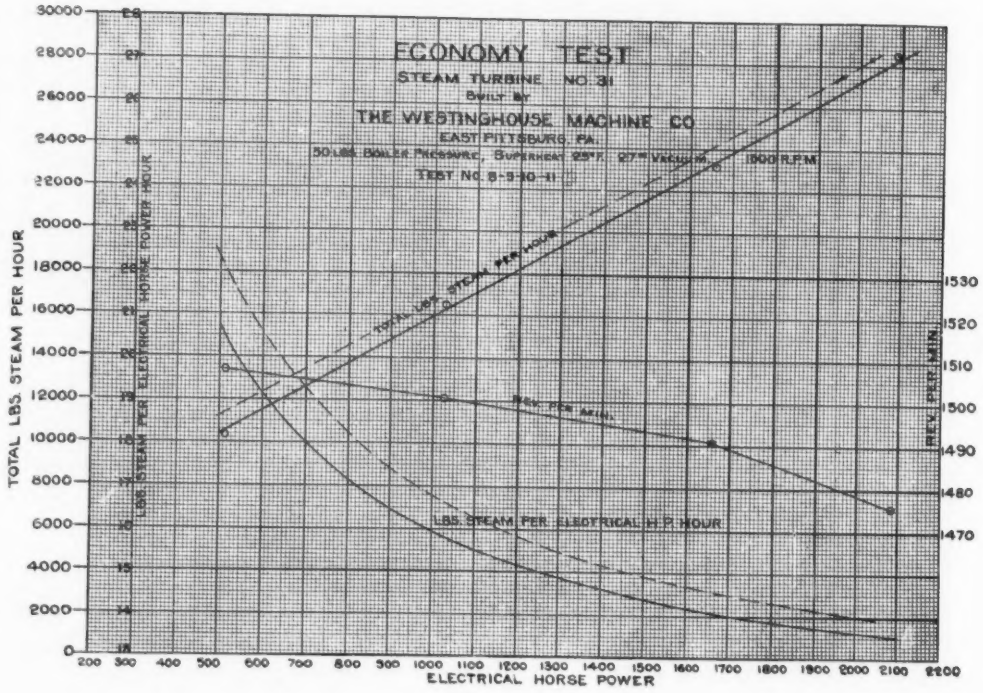

ECONOMY TEST
STEAM TURBINE NO. 31
Built by
THE WESTINGHOUSE MACHINE CO
EAST PITTSBURG, PA.
50 LBS. BOILER PRESSURE, SUPERHEAT 25°F. 27" VACUUM
TEST NO. 8-9-10-11
1800 R.P.M.
TOTAL LBS. STEAM PER HOUR
LBS. STEAM PER ELECTRICAL HORSE POWER HOUR
REV. PER MIN.
LBS. STEAM PER ELECTRICAL H.P. HOUR
ELECTRICAL HORSE POWER
30000
28000
26000
24000
22000
20000
18000
16000
14000
12000
10000
8000
6000
4000
2000
0
200 300 400 500 600 700 800 900 1000 1100 1200 1300 1400 1500 1600 1700 1800 1900 2000 2100 2200
1530
1520
1510
1500
1490
1480
1470

PLATE 2.

Vacuum referred to 30-inch barometer, .	27.5	27.83
Dryness of steam at throttle, . . .	0.994	0.995
Steam per hour per electrical H.P., . .	13.99	15.31
Steam per hour per brake H.P. delivered at the turbine coupling at 95 per cent. generator efficiency, ...	13.29	14.54
Steam per hour per indicated H.P. of a reciprocating engine at 85 per cent. combined efficiency of generator and engine,	11.9	13.01

24. The dotted-line curves in the diagram, Plate 2, indicate the steam consumption when dry saturated steam was used, while the full-line curves answer to a superheat of 25 degrees Fahrenheit of the steam.

25. The advantages of the steam turbine as a motor for propulsive purposes in naval vessels not yet having been fully demonstrated, it seems reasonable, considering its efficiency when used for other purposes, to conclude, that some experiments and actual installations in one or two types of vessels of the Navy are fully warranted.

Very respectfully,

A. B. CANAGA,
Commander, U. S. N.
ERNEST N. JANSON,
Of the Bureau of Steam Engineering, Navy Dept.

The CHIEF OF THE BUREAU OF STEAM ENGINEERING.

NOTES.

TURBINE ENGINES FOR AN ATLANTIC LINER.

Great interest has been excited among ship-building circles in Belfast and elsewhere by the decision of Messrs. Allen, of the Allen line, to fit their new steamer with turbine engines. Messrs. Allen have been considering to some considerable extent the question of turbine propulsion for some time past, and have now decided to take the important step of fitting them into a first-class Atlantic liner. They will thus be the first to lead the way, as far as practical demonstration goes. The contract for the new vessel has been placed with Messrs. Workman, Clark & Co., Belfast. She is intended for the Liverpool-Canadian mail service, and she will be the largest of the Allen fleet, and also the fastest. Her length will be over 500 feet, and her gross tonnage about 12,000 tons. The turbines to be fitted on the vessel are intended to develop about 10,000 horsepower, and the rate of speed guaranteed in the contract is 17 knots per hour. This will be two knots per hour faster than any steamer now running under the Allen flag, and the probability is that in actual working a considerably higher speed will be attained. The speed of 17 knots will mean a reduction in the duration of the voyage from Liverpool to Canada of about a full day of twenty-four hours. The ship will be ready in the course of next year, and the question of the success of turbine engines in vessels of this class will, no doubt, be decided before the launching of the new Cunarders. The success which has crowned the Parsons steam turbine on the Clyde and Channel service boats has, doubtless, encouraged and forwarded their trial on oceangoing and passenger ships generally. It is believed that Messrs. Denny & Bros., Dumbarton, intend also to build a turbine vessel, to steam between Australia and New Zealand.—"The Steamship," Nov., 1903.

OBITUARY.

LIEUTENANT COMMANDER MARTIN BEVINGTON, U. S. NAVY.

Lieutenant Commander Martin Bevington died at the home of his mother, in Mansfield, Ohio, October 25, 1903. He entered the Naval Academy September, 1875, and graduated June, 1879, in the second class of Cadet Engineers that completed the four years' engineering curriculum at that institution. He was promoted to Assistant Engineer June, 1881, and to Past Assistant Engineer June, 1892.

From 1879 until 1893 he served on the North Atlantic, South Atlantic and Pacific Stations, having been attached to the *Wachusett*, *Pensacola*, *Michigan*, *Kearsarge* and *Boston*. During the same period his shore duty comprised a tour of duty of three years at the Bureau of Steam Engineering and three years as inspector of material at the Homestead Steel Works.

In 1893 he was ordered to duty at the works of William Cramp & Sons, and remained there until March, 1894, as assistant to the chief engineer in fitting out the *Columbia*. He was detailed as first assistant to the chief engineer when the *Columbia* went into commission, and remained on that vessel until 1897, during which period the *Columbia* made her remarkable run from the Needles, England, to Sandy Hook, New York—a distance of over 3,000 miles—in less than seven days—a run across the Atlantic that has never been equaled by a man-of-war. He served three years in that vessel.

In 1897 he was ordered to duty in the Bureau of Steam Engineering, Navy Department, and was detailed to organize the inspection of material for that Bureau. His intimate knowledge of the details of this work, combined with his executive ability, experience and study of the subject, made his selection for this duty a particularly suitable one.

After the enactment of the Personnel law, whereby the duties of Line and Engineer officers were amalgamated, he was ordered to duty for compass instruction, and then to the flagship *New York* for Line duties. His desire and special fitness for engineering duty, and his interest in his original profession, induced him, however, to request engineering duty only; and he was therefore ordered, in December, 1899, to Newport News for duty in connection with fitting out the machinery of the battleship *Kentucky*. He was detailed as senior engineer officer of that vessel when she went into commission in May, 1900, and served as her chief engineer until April of the present year, when he was ordered home. During the latter part of his service in Asiatic waters he was fleet engineer of that squadron. In March, 1902, he was promoted to Lieutenant Commander.

Lieutenant Commander Martin Bevington had an excellent presence, and possessed the best attributes that go to make a high type of officer and man. Fearless in maintaining his convictions, there was a kindly side of his nature which endeared him to his many friends and associates. His professional attainments were acknowledged and testified to by all superior officers under whom he served. The esteem and affection in which he was held was evidenced by the profound sorrow of the many, within and without the Service, who deplored his untimely death.

He was a member of many clubs, among which were the Pittsburg Club, New York Club, Naval Order of the United States, and the Metropolitan and Army and Navy Clubs, of Washington. There were few officers in the Navy who were more universally popular and better known in the Service than Lieutenant Commander Bevington.

LIEUTENANT A. M. BEECHER, U. S. NAVY.

The American Society of Naval Engineers and the Naval Service sincerely deplore the death of Lieutenant A. M. Beecher, U. S. N., who lost his life by falling down the turret of the

U. S. S. *Maine*, November 3, while in the performance of official duty.

Lieutenant Beecher was appointed from Iowa to the Naval Academy June, 1880, and graduated with honors from that institution in 1884. He was not only a brilliant officer, but possessed to a remarkable degree the power of application, study and concentration.

In the performance of every duty he was exceedingly thorough and conscientious. Conservative by nature and calm in judgment, he was yet nevertheless prompt in decision and quick in action. His professional career was of an exceedingly varied character. He rendered valuable service in connection with survey work in Alaska. He was a member of the Guam Survey Board, wherein his general information and knowledge was of high value. His last shore duty was at the Bureau of Equipment, where he performed service as Wireless Telegraphy Expert of the Navy.

Lieutenant Beecher possessed a constructive engineering mind in addition to rare capacity for leadership. By reason of his inclination for technical work, he had a special desire to fit himself for important duty of this nature, and engineering interests in the Navy experienced a loss in his death.

The important service that he rendered the Navy is known in its fullness to his confrères at the Bureau of Equipment, and the results attending the special work performed by him at the Department will be evidenced in the more efficient equipment of the armored cruisers and battleships that are about to go into commission or are in process of construction.

By reason of his unassuming and unselfish nature, he was greatly beloved in the Service, and especially was this nobility and gentleness of character recognized by his classmates of 1884. Efficient in the discharge of every duty assigned him, considerate to those serving under him, keeping in touch with his contemporaries, loyal to superiors, and with an earnest love for the Navy, he was a type of officer who, by precept and example, upheld the best and highest traditions of the Service.

NOTICE.

As the Council of the Society has been informed that some of the Members and Associates, as well as several scientific libraries, are desirous of completing their files of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, the Secretary-Treasurer during the present year purchased some special numbers that were offered for sale.

A careful account of stock of various numbers of the JOURNAL now on hand shows that the following numbers can be procured. Since there is a continual demand for copies of special numbers, it would be advisible for Members to write promptly for numbers that are necessary to complete their sets.

JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

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1890	2	0	0	2	1
1891	3	1	19	2	2
1892	4	0	4	0	2
1893	5	2	18	0	4
1894	6	13	16	1	4
1895	7	1	9	13	0
1896	8	2	2	0	13
1897	9	1	0	1	12
1898	10	1	16	5	3
1899	11	56	19	11	11
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